A REVIEW OF LITERATURE ON DYNAMIC MODELS OF VAPOR COMPRESSION EQUIPMENT

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1 Introduction

1.1 Objectives

Existing literature consists of significant work on vapor compression system models; however, not as much exists on transient models of complete heat-pump systems and even less on chiller systems, which constitute a sizeable portion of the commercial refrigeration equipment in use. This report attempts to review the existing literature on dynamic models for vapor compression systems, components and controls and summarize, among other things, the methodologies adopted, and their applicability to chiller systems. The review includes papers with different approaches to transient modeling of individual components as well as complete systems.

1.2 Sources

The papers reviewed here are taken from many sources, primarily ASHRAE Transactions, International Journal of HVAC&R Research, International Journal of Refrigeration, Applied Thermal Engineering, International Journal of Energy Research, IEEE Transactions on Power Delivery, ASME Journal of Dynamic Systems, Measurement and Control, International Journal of Multiphase Flow, Energy Conversion Management, Conference Proceedings of the 15th and 20th International Congress of Refrigeration, National Bureau of Standards and graduate theses. A valuable resource for this compilation was the Reference Guide for Dynamic Models of HVAC Equipment by Jean-Pascal Bourdouxhe and Jean Lebrun, for ASHRAE. The literature reviewed, spans about 23 years, beginning with Wedekind et al [1978] to the ASHRAE Meetings of 2000 and 2001.

2 Organization

This section begins with an overview of transient modeling identifying the phenomenological issues involved followed by the common simplifying assumptions made and the generic approaches to building models. The literature reviewed, is segregated in the following manner, based on the primary focus:

Complete vapor compression systems

Sub-systems or components

Applications

Each of the papers was reviewed using the following general template:

Equipment Description

This gives a quick look at the specific equipment that is modeled and also identifies the primary focus of the work.

Purpose *(where appropriate)*

The purpose for which the model was developed is stated, wherever the objective behind the work was not immediately apparent.

Assumptions

This lists the simplifying assumptions made in the model.

Mathematical Description

This presents a brief description of the mathematical formulation of the model and how it is arrived at.

Solution Technique

This identifies the method in which the solution of the mathematical form was executed.

Applicability

The generality of the model is assessed here and commented on.

Discussion

<u>Validation</u>: Describes how the model was validated and comments are included about significant conclusions drawn.

<u>Comments</u>: These are included wherever it was felt that some specifics of the work needed to be highlighted or some concerns expressed.

References

This consists of a shortened list of the references, which are considered critical in getting a complete picture of the work presented.

2.1 Overview of Transient Modeling

In theory, the complete operation cycle of a refrigeration system can be characterized by two major time-regimes, namely transient and steady state. In the latter, the system input/output parameters are constant over time; transient operation is then, by default, the non-steady state. Typically, this is the case when the system is started-up and is approaching steady state, or when it is shutdown from a steady state, or when it is disturbed from its steady state. This disturbance could be caused by either external changes in conditions (such as load, ambient temperatures etc.) or by feedback control. In either case, the system attempts to move from one equilibrium state to another. Transient modeling is the predictive analysis of the system's operation *during* such conditions. In practice, however, there exists a third time-regime in between the true steady state and transient perspectives, termed the 'quasi-steady state', in which the systems transient responses are much faster than the transients of the inputs. What this means is that the system moves quickly through a sequence of steady states, even while subjected to time varying conditions. This representation is useful when the time constants of the inputs and of the system differ by orders of magnitude. For such cases, steady-state modeling could be used to study transient behavior.

The process of building a system transient model consists primarily of building transient models of the individual components and then integrating them into a whole system. For the system model to truly represent actual behavior, each of the components must in themselves be accurate. This requires a sound understanding of the phenomena occurring in these components and the ability to mathematically represent these phenomena to the degree of accuracy needed.

A pre-requisite for building a mathematical model is the identification of the *scale* of the transients of interest. Broadly, the transients of a system could be categorized as small scale and large scale, based on the relative time-constants of the responses and their causes. Large-scale transients are caused by load changes, start-up, shutdown, feedback control, etc and the responses are scaled on the same order-of-magnitude as the total cycle time. Small-scale transients are caused by (possibly random) fluctuations in conditions, e.g. compressor valve dynamics. The responses in this case are scaled on a much smaller time scale.

2.2 Model Requirements

During transient operation, all the components experience phenomena absent in steady state operation, due to the non-uniformity of conditions within them. The refrigerant mass flow rate, in general, is continuously changing, causing changes in refrigerant distribution in the system components, inlet/outlet conditions of the compressor and the operating point of the expansion device. Of the four major components in the vapor compression system, the transients in the heat exchangers are usually the slowest and have the largest impact on transient performance. It is necessary to consider mass distribution within the heat exchangers as a function of time and space and this requires transient mass balances to allow for local storage. Thermal capacitances of the heat exchanger bodies and the refrigerant have to be considered to account for local energy storage. When the secondary fluid is a liquid such as brine or water, the thermal inertia of this fluid also becomes a significant factor. To determine spatial and time variations of pressure within the heat exchangers, which is the driving potential for mass flow, the transient form of the momentum balance has to be used in some form. This is particularly complicated since it requires solution of the Navier-Stokes equations for compressible flow.

Other phenomena that may be considered are oil migration and foaming in hermetic compressors, liquid slugging into the compressor, flashing in the expansion device, hunting etc.

In addition to predicting the important phenomena, the transient model should be fast enough to be practical. This requires the identification of suitable assumptions that can simplify the mathematical form without loss of relevant detail. Efficient numerical techniques are also necessary to reduce the computation time, thereby allowing the model to run as close to real-time as is possible, while also constraining errors to acceptable limits.

In general, a transient model is a set of coupled space-time partial differential equations in mass, energy and momentum balances, which evolve to more manageable ordinary differential and algebraic equations when simplifying assumptions are applied.

2.3 Common Simplifying Assumptions

Some of the common assumptions found to be made (though not all simultaneously) in the models studied in this literature review were:

- Flow in the heat exchangers is one dimensional and homogenous
- Axial conduction in the refrigerant is negligible
- Liquid and vapor refrigerant in the heat exchangers are in thermal equilibrium
- Effects of pressure wave dynamics are negligible
- Expansion is isenthalpic
- Compression is isentropic or polytropic
- Thermal resistances of metallic elements in the system are negligible in comparison with their capacitances.

2.4 Modeling Approaches

In general, a component may be modeled either from a known (manufacturers) map of its performance or from basic engineering principles. Use of component maps keeps the model close to known performance and makes it easier to keep track of the uncertainty. The flip side of using maps is that the model is applicable only within the range in which the map was generated, and extrapolation is risky. The first principles approach, on the other hand, is inherently more robust but also more expensive computationally.

From the literature reviewed it was observed that maps, when used, were adopted only for the compressor. Invariably, the heat exchangers and expansion devices were modeled from first principles. Also, it was seen that the largest task in modeling a refrigeration system was, usually, the modeling of the heat exchangers.

For heat exchangers, a sub-classification of modeling techniques was found. These are the phase-dependent moving boundary method and the phase-independent finite difference methods. In the former approach, the heat exchanger is divided into sections of variable volume according to the state of the refrigerant, i.e. liquid, two-phase or superheated. Since there is constant mass redistribution during transient operation, these volumes cannot be constant and it is necessary to track boundaries between adjacent volumes, which ‘move’ within the heat exchanger. In the condenser this could lead to tracking two boundaries – one between the superheated vapor and the two-phase regions, and the other between the two-phase and the sub-cooled regions. Special attention needs to be given to situations when the phase boundary moves in or out of the heat exchanger, such as happens during large scale transients. For example, when a system is started up from a condition where it is in equilibrium with the ambient, the condenser typically contains only superheated vapor. Subsequently, as the compressor begins to “lift” the refrigerant from the evaporator to the condenser, condensation begins and a two-phase region develops. Further along in time, when sufficient refrigerant has been moved to the condenser, a sub-cooled, liquid region forms. In the finite difference approach, the governing conservation equations are approximated by a finite difference scheme that typically consists of dividing the heat exchangers into a number of (possibly constant volume) elements, and each element is defined with its own state properties. The formulation for any element is phase-independent and therefore identical in all three phases. A third approach is to combine the moving boundary with finite differencing. This involves a phase-dependent finite differencing with elements whose volumes change over time with no elements spanning phase boundaries.

Another sub-classification of modeling techniques is in the choice of a lumped versus distributed parameter method. The lumped parameter method is computationally simpler, since it results in a finite system of equations consisting of algebraic and 1st order, ordinary differential equations. However, the drawback is spatial detail is lost by averaging the state parameters over the complete control volume. The distributed method approach allows spatial variations to be monitored and is simpler in that the governing equations for all the elements are identical, if the property relations can be suitably framed to be phase-independent. The level of spatial detail depends on the level of discretization. The trade-off with this approach is the much larger computational time required and greater attention to potential numerical instabilities.

In addition to the above classifications, the flow of two-phase refrigerant can be modeled either using a homogenous or a slip-flow model. In the homogenous model, the liquid and vapor phases are considered to be in thermal equilibrium and moving at the same velocity. In the slip flow model, the liquid and vapor phase velocities are different and the model needs to be built based on the nature of the flow, i.e. bubbly, slug, annular, etc. In such models, it is not necessary that the liquid and vapor phases would be in thermal equilibrium at any section.

3 Overview of the Literature

Wedekind et al [1978] were among the earliest to study transient behavior with their work on modeling of two-phase flow dynamics in heat exchangers. In an attempt to simplify representation of two-phase flow, their model is built from a moving boundary formulation using a variable volume form of the volumetric mean void fraction over the two-phase region. A significant achievement of this is that the complete two-phase region can be treated in adequate detail even in a lumped form, while avoiding the necessity of handling the transient form of the momentum equation.

Dhar and Soedel [1979] present one of the first models of a complete vapor compression refrigeration system. This model of a window air conditioner is built from first principles using a moving boundary approach in the heat exchangers. Two-phase refrigerant in the heat exchangers is treated as a coupled pair of lumps representing the liquid and vapor phases that exchange mass internally and heat externally. The model also accounts for refrigerant dissolution in the oil of the hermetic, reciprocating compressor. Despite using fairly simple representations, all major transients are well captured. The development focuses on the refrigerant side leaving the secondary fluid open to choice. This could arguably be used on liquid chillers with the selection of a suitable compressor model.

Goldschmidt and Hart [1982] developed a lumped model for a refrigeration system with a view to study its seasonal performance when coupled to a mobile home whose heat losses were modeled in detail. The focus of this work being on the dynamics of the residence itself, the air-to-air heat pump system itself is modeled using an exponential fit to steady state performance. The steady state performance of the heat pump is regressed from manufacturer's data and experimental measurements.

Chi and Didion's [1982] model is among the few that works with the transient form of the momentum equation. Their model of an air-to-air heat pump system, is built on a moving boundary lumped parameter formulation. The start-up transients when operating the model in cooling mode are analyzed. All the components, including the heat exchanger fans and the motor shaft are included. The dynamics of all components are captured, including the momentum of air flowing across the heat exchangers and rotational inertia of the motor shaft. Dynamics of the valve however, are not considered significant.

Yasuda et al [1983] built a complete system model, on lines similar to Dhar and Soedel [1978], except that the condenser modeled is a shell-and-tube construction instead of air-cooled. The object of this model is to capture what are termed *small* transients. These include transients caused by feedback control and by instability triggered by poor valve setting. The purpose of this model being narrowly defined, broad assumptions such as constant sub-cooling and uniform two-phase condition in the entire condenser, are found to work without serious consequence.

MacArthur [1984a] presents one of the earliest of models that moves away from the lumped parameter approach towards a distributed formulation. This, along with McArthur and Grald [1987] and Rasmussen et al [1987], constitute a body of work using similar formulations for the system components. The space-time dependent conservation equations are simplified by assuming one-dimensional flow in both heat exchangers. The two-phase region in the condenser is treated as homogenous whereas in the evaporator the liquid and vapor phases are modeled separately. The [1984] work uses a simpler version of the heat exchanger formulation in that, the pressure response of the heat exchangers is de-coupled from the thermal response by the imposition of uniform flow velocities along the heat exchanger length. This yielded inaccurate mass distribution predictions and the issue is addressed in McArthur and Grald [1987] where the mass balance is coupled to the energy balance and allowed to dictate the pressure response. In all of these, the heat-exchanger's discretizations were fully implicit thereby allowing stable solutions for time steps up to 10s. Rasmussen et al [1987] build on this refined model, and include the compressor's prime mover, specifically an engine. The important thermal and inertial dynamics of the engine are modeled and coupled to the vapor compression heat pump system model.

Murphy and Goldschmidt [1984, 1985] developed simplified system models to study start-up and shutdown transients for an air-to-air system. In the start-up model, the dynamics dealt with are those of the capillary tube and the phenomenon of liquid backing up into the condenser during start up. The compressor is modeled from steady-state measurements and actual measurements of evaporator performance are used in place of an evaporator model. The condenser dynamics modeled are those of the refrigerant pressure response and the tube material. In the shutdown study, both the heat exchangers are modeled as tanks containing two-phase refrigerant at different pressures to begin with, with air as the secondary fluid cooling or heating the coils by natural convection. The refrigerant is allowed to flow only through the capillary tube initially and subsequently also through a pressure equalization valve that opens shortly after shutdown begins. The liquid line is modeled in detail to capture the variation in refrigerant quality between entry and exit so as to ensure an accurate entry condition to the capillary tube which predominantly controls the shutdown process.

Sami et al [1987] used a lumped parameter approach to model the system components where the dynamics were relevant. Multiple component configurations are modeled including shell and tube condenser and evaporator, air-cooled condenser, direct expansion evaporator, capillary tube and thermostatic expansion valve. The model of the hermetically sealed reciprocating compressor is taken from Yasuda et al [1983] and enhanced by including oil dissolution in the refrigerant. The heat exchangers are modeled using a drift-flux model that consists of separating the vapor and liquid phases and coupling the mass and energy balances of the individual phases through the evaporation or condensation mass and energy exchange. This model is built to allow cooling and heating cycle operation of the system. Validation of the model is provided for the start-up performance of a liquid chiller.

A unique case of a ground-coupled heat pump system was developed by Safemazandarani [1990]. A transient model is constructed of the heat exchanger coupled to the ground and the system's response in cooling and heating modes is analyzed. As an improvement to the direct coupling of the heat exchanger with the ground, a proposal is analyzed in which a water bath is introduced between the heat exchanger and the ground.

This proposal was found to require a smaller heat exchanger because of the improved heat transfer characteristics with the water.

Nyers & Stoyan's [1994] model of an evaporator is built on the moving boundary formulation using finite differencing within each phase. This model has been used to predict the evaporator's behavior under step jump, exponential saturation, and periodic oscillation of the temperature and flow rate of the secondary fluid, compressor speed, condenser pressure, and throttle coefficient.

Vargas & Parise [1995] studied the relative benefits of an alternative form of closed-loop control method, based on a power-law, against the conventional on-off cycling. Their model is a highly simplified, component level lumped parameter formulation, used to highlight the improvements in cyclic energy efficiency when the proposed closed-loop control method is used.

Sami & Comeau [1992] and Sami &Dahmani [1996] expanded on the model of Sami et al [1987] to include finite differencing within the drift-flux model. This model was used to predict system performance. The [1992] work dealt with non-azeotropic refrigerant mixtures, specifically mixtures of R22-R114, R22-R114a, R22-R152a, while the [1996] work dealt with HFC alternatives to R22, specifically, R407a, R507 and NARM502 (a blend of R22, R23 and R152a).

Ploug-Sørensen et al [1997] constructed a model of a domestic refrigerator, using the software package SINDA/FLUINT to demonstrate the capabilities of this package which is used extensively in the aerospace industry. Using the domestic refrigerator as an example, the many features and mathematical infrastructure of the package are explained.

Xiandong He et al [1997] developed a system model for a basic vapor compression refrigeration system, using the moving boundary lumped parameter formulation with the system mean void fraction method of Wedekind [1978], for the purpose of studying the effect of multivariable feedback control. This model was then used to study a multi-input-multi-output (MIMO) control method developed by Xiandong He et al [1998].

Williatzen et al [1998] present a model for simulating the transient flow dynamics in a heat exchanger, in the form of a set of lumped parameter moving boundary formulations. The structure of the model allows for any physically possible combination of phases within the heat exchanger to be handled by an algorithm which switches between the appropriate sets of equations. Pettit et al [1998] applied this formulation to the case of an evaporator and studied the phenomena of the appearance and disappearance of phases-regions within the evaporator.

Rossi & Braun [1999] developed a fast yet largely mechanistic model of a roof-top air conditioning unit. The importance of real time simulation is emphasized, and a smart automatic integration step sizing algorithm is presented that robustly simulates start-up and on-off cycling. The system model is constructed using a fully finite-volume formulation of the mass and energy balances in the heat exchangers. Validation is presented using start up measurements of a 3-ton roof top unit.

Jing Xia et al [1999] developed a separated flow model for an evaporator and studied the dynamics of the evaporator under variations of compressor speed, condensing temperature, secondary fluid conditions and expansion valve opening.

Jakobsen et al [1999] analyzed the relative accuracies of assuming homogenous flow and slip-flow patterns in the heat exchanger, and concluded that the homogenous flow model was an inadequate representation, which over-predicted the sensitivity of the evaporator. They recommend the use of the slip-flow model when the dynamics of the refrigerant are of interest.

Svensson [1999] was one of the few investigators to focus exclusively on liquid chillers. This model is constructed on a phase-wise lumped parameter formulation in the heat exchangers, which is adequate for the study of the dynamics of the system to load disturbances. These disturbances are introduced by step changes in the condenser side water flow rates.

Mechanistic, single stage and two-stage centrifugal chiller models are presented in Wang and Wang [2000]. A detailed model of the centrifugal compressor is developed from first principles, i.e. the momentum equation, the energy equation and the velocity triangles. All major losses namely hydrodynamic, mechanical and electrical are accounted for. Also incorporated is the inlet guide vane or pre-rotation vanes form of capacity control. The heat exchangers, however, are modeled in a highly simplified manner. Both are treated as single entity lumps with overall conductances and effectivenesses correlated to the water flow rate and constructional parameters. The dynamics of the chiller are introduced in the form of lumped thermal capacitances to account for the refrigerant, heat exchanger body and secondary fluid (water in this case).

Browne and Bansal [2000] developed and compared a simple physics based dynamic model with a dynamic neural network model of a screw chiller system. The screw compressors are modeled as steady state devices incorporating capacity control in the form of varying the swept volume in response to the error in the chilled water temperature. The dynamics of the system are in the form of lumped elements for the heat exchangers material and the water. The refrigerant in the heat-exchangers is treated quasi-statically.

Grace I.N. and Tassou S.A. [2000] developed a simplified dynamic model of a liquid chiller consisting of a reciprocating compressor and shell-tube heat exchangers. The condenser has the refrigerant flowing in the shell and the evaporator has the refrigerant flowing in the tubes. The discretization and solution of the heat exchangers is identical to that developed by MacArthur and Grald [1987]. The expansion device is a thermostatic valve with a sensing bulb. The expansion is modeled as isenthalpic. The superheat sensing bulb is modeled in detail by accounting for all the relevant heat transfer resistances and capacitances.

4 Conclusions

Dynamic performance of vapor compression systems has been of interest for well over 20 years, from Dhar [1978] and through to Browne and Bansal [2000]. As can be seen from the history of work on the topic, interest appears to be growing in recent years on development of better and more detailed dynamic models. Also seen is a growing interest in chiller systems, specifically liquid chillers, as opposed to the preponderance of air-to-air system studies in earlier years.

Within the scope of the literature gathered for this review, a short-listing of liquid chiller models yielded Sami et al [1987], Svensson [1999], Wang and Wang [2000] and Browne and Bansal [2000]. On examining these models critically, the absence was noticed of a single dynamic system model of a vapor compression centrifugal liquid chiller that included detailed heat exchangers and which could adequately model large and small scale transients. Sami's model was limited to a hermetically sealed reciprocating compressor, while Browne's dealt with screw compressors only. Svensson's work focused only on transients triggered by feedback control. Wang's model, which came closest to the above search criteria in that it was a centrifugal liquid chiller model that could capture load changes, fell short in its over-simplification of the heat exchanger dynamics. In consonance with the findings in this literature review, Browne and Bansal in their compilation work on issues related to modeling of vapor compression liquid chillers highlight the need for a liquid chiller model that incorporates detailed heat exchangers.

In conclusion, a large volume of literature related to dynamic modeling of vapor compression equipment and spanning over 20 years, was reviewed, and it was found that no system models existed that could predict the complete dynamic performance of centrifugal liquid chillers despite such systems being among the more popular configurations in the field.

5 Dynamic Models

5.1 Complete vapor compression systems

5.1.1 Transient Analysis Of A Vapor Compression Refrigeration System

Part I - The Mathematical Model & Part II - Computer Simulation and Results
M. Dhar and W. Soedel, 1979
Proc. 15^{th} International Congress of Refrigeration (Venezia)

Equipment Description:

A complete vapor compression system, in air-conditioning application, is presented. The development focuses on the refrigerant side of the system, leaving the secondary fluid open to choice. The compressor is a hermetically sealed, reciprocating machine. An accumulator is included between the evaporator and the compressor shell. Refrigerant dissolution in the compressor oil is also modeled. Two valve designs, thermostatic expansion and pressure regulator, are modeled.

Purpose:

This model was developed for the purposes of predicting compressor behavior during start-up, focusing on identifying conditions that could lead to liquid slugging, and also as an aid to the system designer in determining optimal performance.

Assumptions:

- Compression is polytropic.
- Valve pressure drops are neglected.
- Internal resistances of metallic components are neglected.

Mathematical Description (from Part-I):

In addition to the dynamics of the refrigerant itself, capacitances of the heat exchanger wall material, compressor shell material, compressor cylinder and piston mass, lubricating oil mass in the compressor shell, refrigerant resident in the accumulator and mass of the valves' sensing bulb are modeled.

The model is constructed on a moving boundary formulation. Heat exchangers are treated to behave in modes determined by the state of the refrigerant within them. The condenser operates either in a fully superheated condition or in combined superheated and two-phase or as superheated, two-phase and sub-cooled. The evaporator is treated a little differently, in that it operates in one mode only, which consists of a liquid zone and a vapor zone. Unlike in the condenser where in the two-phase region the liquid and vapor are considered to be in thermal equilibrium, in the evaporator, the liquid and vapor phases exchange heat. This allows for the prediction of non-zero superheat.

Within the compressor, the zones of interest are the refrigerant volume in the shell, the variable volumes of the lubricating oil at the base of the shell and refrigerant volume within the cylinder and the thermal masses of the cylinder-piston arrangement and the compressor shell.

Application of relevant mass and energy balances to each of the identified zones yield a system of closed, coupled algebraic and first order differential equations. Each of the zones is treated as a fully mixed region with exit conditions being equal to the bulk mean conditions within. In order to simplify the computation, pressure drops within the heat exchangers are considered negligible while solving the system cycle. This eliminates the need for the momentum conservation equation. However, steady state pressure drops in the heat exchangers are computed after solving the system, and the pressures are appropriately adjusted.

For a detailed description of the formulation, the reader is referred to the author's doctoral thesis Dhar [1978].

Solution Technique (from Part-II):

The system of algebraic and 1st order ODEs developed in Part I of this work are solved simultaneously, one system component at a time. To quote, "One cycle of operation of the system consists of calling each component subroutine in sequence, and the equations of one subroutine after another are evaluated until all the equations have been solved once". The ODEs are solved by the Euler method, in the interest of minimizing computation time. The integration time step is determined by interval halving, beginning with an arbitrary time-step, and halving it until successive solutions agree to the desired accuracy.

Application of the model:

The model, as mentioned earlier, focuses on the refrigerant side of the system, and therefore allows for use with different heat exchangers. Appropriate code would need to be developed to model the secondary fluid. This indicates that the model can be used for heat exchangers that are air-cooled or of shell-and-tube construction. Also, the choice of refrigerant is left open. Although no comment is made as to how the properties are computed in this model, it is believed that alternative refrigerants can be ported to the system. The compressor and expansion device, however, are constrained to be hermetic and thermostatic, respectively. However, the modular nature of the model coding allows for mix-and-match arrangements of other expansion devices and compressor models.

Discussion:

<u>Validation:</u>

The model was validated against a window air-conditioner rated at 2.285kW. Several transients were independently studied:

- using a thermostatic expansion valve
    - start-up, keeping the room conditions constant,
    - start-up with the room temperature varying,
    - start-up and an off-cycle triggered by falling room temperature,
- using a pressure regulator
    - start-up, keeping the room conditions constant,
    - start-up, with the room temperature varying,
- start-up through steady-state, followed by a drop in the outside ambient temperature (not clear which valve arrangement was used)

In addition to the above transient performance, an additional case of compressor abuse was studied by insulating the compressor. All the important transients in the system were well predicted.

Comments:

This model is the earliest system transient model found in this literature search. The formulation is simple, yet captures all the major dynamics of the system. The system validation is exhaustive and covers most major transient phenomena relevant to an air-conditioning system. Feedback control, however, is not built into the model, and therefore not presented.

References:

Dhar M., 1978, "Transient Analysis of Refrigeration Systems" - Ph.D. Thesis, Ray. W. Herrick Laboratories, School of Mechanical Engg., Purdue University.

Soedel W., 1972, "Introduction to Computer Simulation of Positive Displacement Type Compressors" - Short Course Notes, Ray W. Herrick Laboratories, Purdue University

5.1.2 Heat Pump System Performance: Experimental And Theoretical Results

V.W. Goldschmidt & G.H. Hart, 1982
ASHRAE Transactions Vol. 88, Part 1, pp. 479-489

Equipment Description:

A model of a heat pump coupled to a mobile home is presented. The development focuses on the heat load and losses in the home, and simplifies the heat pump itself to a single lump.

Purpose

The purpose of the model is to predict heat loads in a residential heated space during daily and seasonal changes in ambient conditions. This prediction is coupled to a simplified heat pump model.

Assumptions:

The complete heat pump being modeled as a single lumped system, the only significant assumption in its model is that the transient capacity is exponentially dependant on its steady state performance.

Mathematical Description:

Simple relations are developed from measurements for the following:

- fractional on-time of the heat-pump system, in terms of the outside DBT,
- increase in on-time caused by defrosting cycles, in terms of the outside humidity
- steady-state heat capacity of the heat pump in terms of the outside DBT,
- transient heat capacity of the heat pump as an exponential of its steady-state capacity

The steady-state capacity of the system is taken from the manufacturer's maps for the systems performance. The rest of the model centers on determining the cycling times for the heat pump, and using these cycling times in the exponential map to determine the transient heat capacity. The heat pump model takes as inputs, the outside weather conditions and predicts, as outputs, the heat capacity, power consumption and COP.

The model of the home itself, is a steady-state one and takes, as the input, the outside weather conditions, and predicts the heat load in the home, accounting for duct and infiltration losses. It is executed on an hourly basis. Also accounted for is the on and off cycling that is triggered by the formation of frost.

From the outside dry-bulb temperature, the steady-state thermal capacity of the heat pump is determined from a regression based on measurements and calibrated against the manufacturer's map. The transient capacity of the system is computed based on an approximation that it approaches the steady-state capacity exponentially in time. From steady state and overall cycle energy balances between the heat pump and the home, the fractional cycle on-time is related to a minimum cycle on-time. This minimum on-time is pre-determined, as a constant, in terms of the thermostat accuracy and the thermal mass of the building. This suffices to determine the cycling time of the heat pump. In conjunction with the steady-state model of the heat pump, the transient capacity is determined. Based on outside humidity and the evaporator coil surface, the possibility of frost formation is predicted. The defrosting is not modeled directly, but its effect is captured by the fact that the system's on-off cycling times change when frost forms and the system's defrosting mechanism starts operating.

Solution Technique:

The simple and sequential nature of all the correlations developed for this model makes for straightforward solution of the equations.

Application of the model:

Since the focus of this work is on modeling heat losses in the residential space, the heat pump model is simplified to a level that generates only as much information as is necessary for the purpose. Needless to say, this makes for a very application-specific model. Absence of any component detail precludes use of this to study heat pump behavior in any significant form.

Discussion:

Validation:

The models for the heat pump and the home were validated against an existing mobile home, coupled to heat pump rated at 10.55kW. The validation consisted of predicting the home conditions under different outside weather conditions, and then collecting data under such conditions. The model predictions were found to be accurate to within 10%.

Comments:

The model presented in this work is primarily that of a residential space. The heat pump model used is a highly simplified one that treats the entire system as a single lump. No details are presented as to the nature of the heat exchangers or the compressor because they're not modeled individually.

References:

Hart G.H., 1978, "The performance of the air distribution system and of an air-to-air heat pump, in the heating mode, on a mobile home." MS Thesis, School of Mechanical Engineering, Purdue University.

Sanchez C.A., 1975, "The measurement and modeling of heat gains and losses in a mobile home.", MS Thesis, School of Mechanical Engineering, Purdue University.

White R.R., 1975, "Seasonal performance measurement and modeling of a mobile home gas-fired furnace.", MS Thesis, School of Mechanical Engineering, Purdue University.

Murphy W.E., 1977, "Mobile home air-conditioning: an analysis of seasonal performance.", MS Thesis, School of Mechanical Engineering, Purdue University.

5.1.3 A Simulation Model Of A Heat Pump's Transient Performance

Joseph Chi & David Didion, 1982

International Journal of Refrigeration, Vol. 5 No. 3, pp. 176-184

Equipment Description:

The equipment modeled here is a complete vapor compression refrigeration system operating in the cooling mode. The system consists of an air-cooled condenser, a direct expansion evaporator, an accumulator, a hermetic compressor, a thermostatic expansion valve and the heat exchanger fans.

Purpose

This model was developed as a performance evaluation tool for an air-to-air refrigeration system that can be used to predict detailed system parameters.

Assumptions:

- Refrigerant flow in both the heat exchangers is 1D homogenous.
- Compression is polytropic
- Air is treated as an ideal gas.
- Heat exchanger wall thermal resistance is negligible.

Mathematical Description:

The dynamics included in this model are those associated with the refrigerant, the air flowing across the heat exchangers, the fans driving the air, the heat exchangers walls, the motor shaft, the compressor, the expansion valve and the accumulator.

The heat exchangers are modeled with moving refrigerant phase boundaries, with each phase region treated as a counter-flow heat exchanger. Refrigerant properties are computed using equations of state reported by Martin [1959]. Single-phase heat transfer coefficients are used from standard literature McAdams [1954] and Rohsenow et al [1961]. Two-phase flow pressure drop coefficients used are the Lockhart and Martinelli [1949] correlation for condensation and Pierre's [1964] correlation for evaporation. Two-phase heat transfer coefficients are taken from Traviss et al [1971] for convection condensation inside tubes and Chaddock et al [1966] for evaporation in horizontal tubes.

Application of the transient mass, energy and momentum conservation equations yields a system of 1^{st} order differential equations with the state variables defined for each phase region of the heat exchanger being the inlet pressure, leaving mass flow rate and temperature of the air, the density, leaving mass flow rate and state of the refrigerant and the temperature of the tube wall. The inputs are the flow rate, temperature and humidity ratio of the air at inlet, the air pressure at outlet, the refrigerant flow rate and enthalpy at inlet and the refrigerant pressure at outlet. The outputs are the air pressure at inlet, the flow rate, temperature and humidity ratio of the air at the outlet, the refrigerant flow rate and enthalpy at outlet and the refrigerant pressure at the inlet.

Motor shaft dynamics are modeled from an angular momentum balance between the driving and braking torques. The torque-speed characteristics of the motor itself are obtained from manufacturer's data. From these, the driving torque and speed to the compressor are known. The shaft speed, in addition to the refrigerant inlet flow rate, inlet enthalpy and the outlet pressure are taken as inputs to predict the refrigerant outlet flow rate, outlet enthalpy and inlet pressure, along with the braking torque. Within the compressor, suction and discharge head losses are accounted for by linear coefficients proportional to the respective dynamic heads. The compression process efficiency is determined using a compression index n, and the isentropic efficiency. Determination of this index is not detailed. The braking torque is then computed from an energy balance across the compressor.

In addition to the above components, auxiliary components such as the fans, the accumulator and the expansion valve are also modeled. The flow-rates and temperatures of the air supplied by the fans are determined from simple exponentials in the steady-state flow rates and temperatures and energy balances across the fans.

The accumulator is modeled as a lumped heat exchanger, taking inputs of inlet refrigerant flow rate, inlet enthalpy and outlet pressure and predicting the outlet refrigerant flow rate, outlet enthalpy and inlet pressure. The exit enthalpy is constrained to a saturated vapor condition when the accumulator is in two-phase. The leaving flow rate is calculated from an orifice equation for vapor flow.

Lastly, the expansion valve is modeled without any dynamics, computing the refrigerant flow area as proportional to the superheat in excess of the minimum preset. The flow rate itself is computed using an orifice equation.

Solution Technique:

The system of equations described above is compiled to a set of 32 ordinary differential equations that are solved using the explicit Euler integration method with a fixed time step of 0.005s. This value of the time step was determined by executing the model repeatedly, each time halving the step, until successive results agreed to within 0.01%.

Application of the model:

This model was developed for an air-to-air system run by a hermetic compressor. The development is mechanistic in its treatment of the components and can be used to cover a wide range of such systems. The model is coded in FORTRAN and is modular, thereby allowing for alternate constructions and formulations for the individual components to be tried out.

Discussion:

<u>Validation:</u> The model was validated against a 4-ton residential air-to-air heat pump, operating in the cooling mode. The start-up comparison between the model prediction and measurements are presented and show good agreement.

<u>Comments:</u> This model is among the earliest found in our literature review that incorporated a transient momentum balance. The treatment of the major dynamics is detailed. However, no comments are made regarding the controller dynamics, the valve dynamics and the execution speed of the model.

References:

Martin J.J., 1959, "Correlations and equations used in calculating the thermodynamic properties of freon refrigerants" in Thermodynamic and Heat Transfer Properties of Gases, Liquids and Solids, ASME, New York, NY.

McAdams W.H., 1954, "Heat Transmission", McGraw-Hill, New York, NY.

Lockhart, R.W. and Martinelli, R.C., 1949, “Proposed correlation of data for isothermal two-phase, two-component flow in pipes.”, Chemical Engineering Progress, Vol. 45, pp.39-48.
Traviss D.P., Baron A.B. and Rohsenow W.M., 1971, “Forced convection condensation inside tubes,” MIT Heat Transfer Laboratory, Rpt. No. DSR72591-94.
Pierre B., 1964, “Flow resistance with boiling refrigerant,” ASHRAE Journal Pt. 1.
Chaddock J.B. and Noerager J.A., “Evaporation of refrigerant 12 in a horizontal tube with constant wall heat flux.” ASHRAE Trans. Vol. 72, Pt. 1.

5.1.4 Simulation Model Of A Vapor Compression Refrigeration System

H. Yasuda, S. Touber & C.H.M. Machielsen, 1983
ASHRAE Transactions, Vol. 89, Part 2A, pp. 408-425

Equipment Description:

The system modeled here is a complete vapor compression system consisting of a single cylinder open-type reciprocating compressor, a shell and tube condenser, a thermostatic expansion valve and a direct expansion evaporator. Water is the secondary medium in both heat exchangers.

Purpose

This model is intended for systems designers for design optimization and deals with transients in the time-scale associated with feedback control and hunting.

Assumptions:

- Entire condenser is in 2-phase and sub-cooling is constant
- Dynamics of the compressor valves are neglected
- One-directional flow through compressor
- No pressure drop in the condenser
- Vapor in superheat region is incompressible
- Two-phase flow is homogenous and in equilibrium.

Mathematical Description:

The dynamics treated in this model are those associated with all four stages of the (reciprocating) compression process, the refrigerant in the heat exchangers, tube mass, shell mass, water mass and the expansion valve's sensing bulb.

The compressor model is a simplified version of that developed in Touber [1976] & Blankespoor et al [1976], by neglecting valve dynamics. The four processes, namely suction, compression, discharge and expansion are modeled from mass and energy balances, using cylinder pressure and flow-rate as state variables. At the end of each process the cylinder pressure and/or the flow-rate are determined as appropriate. Preset values of valve stiction are used to trigger the switchover from one process to the next.

The condenser is modeled as a combination of lumps to represent the refrigerant vapor, refrigerant liquid, pipe material, secondary fluid (water) and the shell wall. Since the de-superheating region is merged with the condensing, and the sub-cooling is enforced as a constant, the complete condenser is in one (refrigerant) phase. Application of mass and energy balances to each of these lumps results in a system of 1st order, coupled differential equations that require inlet (vapor) flow rate and enthalpy of the refrigerant and outlet (liquid) flow rate of the refrigerant, and the flow rate and entering temperature of the water. This allows the determination of temperatures of the water, tube and shell wall, the enthalpy of the refrigerant and masses of the vapor and liquid refrigerant. Using a mean void fraction, the vapor-liquid distribution is determined and hence the pressure.

The heat transfer coefficients between the refrigerant and the condenser wall and between the condenser wall and ambient air are considered constant, and have been computed from steady-state measurements. The heat transfer coefficient between the refrigerant and the tube is determined from steady-state experimental results and correlated as a function of the refrigerant flow rate.

In modeling the evaporator, the refrigerant in two-phase is treated as a lump, while the superheated region are distributed. The evaporator tube material and secondary fluid are modeled as distributed along both refrigerant phases. This approach is used in order to be able to capture the evaporator exit condition precisely, which, in keeping with the purpose of this model, is important when observing the hunting phenomenon. The lumped two-phase refrigerant region is dealt with on the same lines as in the condenser. The distributed portions of the evaporator are handled using a finite-difference approach. The two-phase and single-phase pressure drops in the evaporator are estimated explicitly, and applied as corrections. Evaporative region mean void fraction is determined from a Hughmark [1962] correlation, assuming uniform heat flux.

In the expansion valve, the only delay considered is that due to the response of the sensing bulb. The dynamics of this bulb are modeled as a series of four lumps, namely, the refrigerant within the bulb, the body of the bulb, the pipe wall to which the bulb is attached and the refrigerant surrounding the bulb. The inertial transients inherent in the motion of the valve are considered negligible in comparison with the system transients. The mass flow rate through the TXV is computed as proportional to the superheat pressure subject to a static superheat pressure.

Solution Technique:

The system of 1st order differential equations developed is integrated using an explicit algorithm. The integration method adopted or the use of any dynamic step-sizing is not reported.

Application of the model:

The model was developed for a shell-tube condenser and a direct expansion evaporator. The formulations adopted do not allow for modeling of the larger scale transients such as start-up, defrosting or cycling and shutdown. The enforcement of constant sub-cooling also precludes fault implementations such as refrigerant over and undercharge.

Discussion:

Validation:

The model was validated against an existing laboratory system with a variable speed compressor motor. The validation exercise was carried out at constant compressor speed and by disturbing (by stepping-up or stepping-down) the static superheat of the expansion valve, when the system is running at steady state. The movement of the system to a new steady state was tracked. The disturbance in the static superheat was carried through to a point of initiating unstable behavior in the system, and the associated performance was tracked. The model was shown to agree well with the measurements for all the disturbances mentioned. The model was also able to capture the system's oscillatory behavior when the static superheat was significantly decreased.

Comments:

This model enforces a strong set of assumptions in the form of constant sub-cooling and uniform two-phase condition in the condenser. For the purposes for which it was developed, these assumptions appear adequate, but the absence of refrigerant inventory restricts the range of transients that can be tracked. The evaporator being a dry-expansion type, it is expected that defrosting is an issue of significance that cannot be handled by this formulation. The concept of separating the refrigerant side pressure drop from the mass and energy balances, and computing it as a post-calculation correction is interesting because it allows the pressure drop to be considered without the computationally expensive solution of the transient momentum equation.

References:

Touber S, 1976, "A contribution to the improvement of compressor valve design", PhD Dissertation, Delft University of Technology.
Blankespoor H.U., Brok S.W., Touber S., 1976, "Simulation of a reciprocating compressor including suction and discharge line by hybrid computer.", Proc. Purdue Compressor Technology Conference, West Lafayette, IN.
Yasuda H., Machielsen C.H.M., Touber S., Brok S.W. & de Bruijn M., 1981, "Simulation of transient behavior of a compression–evaporation refrigeration system.", Delft University Report No. 133.

5.1.5 Analytical Representation Of The Transient Energy Interactions In Vapor Compression Heat Pumps

J.W. MacArthur, 1984

ASHRAE Transactions, Vol. 91, Part 1B, pp. 982-996

Equipment Description:

The complete vapor compression system is modeled, by building component models for refrigerant-in-tube condenser & evaporator, accumulator, fixed orifice and thermostatic expansion devices and a hermetic compressor.

Purpose:

This model was intended for use as a research tool.

Assumptions:

- Uniform, one-dimensional refrigerant flow along the heat exchanger length
- Compression is polytropic

This paper appears to be a precursor to MacArthur and Grald [1987] which deals exclusively with the heat exchangers in essentially the same approach. The present work also includes the other components.

Mathematical Description:

This model is among the earliest forms of a fully distributed representation of the refrigerant dynamics in the system. The dynamics accounted for are those of the refrigerant, the reciprocating compression stages and the thermal capacitances of the heat exchanger material, the secondary fluid, the compressor casing material and accumulator.

The condenser is modeled on a finite-volume scheme by discretizing it along its length into phase-independent control volumes and applying the mass and energy balances on the refrigerant, tube material and secondary fluid. The evaporator on the other hand is modeled differently, in that within the two-phase region the liquid and vapor phases are handled separately while being in mutual thermal equilibrium. Heat from the secondary fluid is directed exclusively to the liquid refrigerant to cause evaporation. Mass and energy balances on the vapor and liquid phases are coupled through the evaporation rate. The single-phase region in the evaporator is modeled in the same way it is done in the condenser.

In the accumulator, mass and energy balances are applied to the liquid and vapor refrigerant, with convective heat transfer to/from both phases. The mass balances of the liquid and vapor phases are coupled by the mass-evaporated term in the accumulator.

The compressor model is essentially a polytropic-efficiency model with suction and discharge states corrected for valve pressure drops and heat transfer to/from the shell and cylinder.

Solution Technique:

The complete system is modeled by combining the individual component models. Plumbing losses are neglected and therefore, the exit state of any component is assumed to be the entry state of the following component.

The solution is executed by a time-wise step through all the components in sequence. At each time step, each component is iterated to equilibrium convergence, based on entry conditions and known-previous component and exit states. At the end of this convergence, updated exit conditions are predicted which are then applied to the component downstream. This sequence is continued until steady state. In solving for the evaporator states, a distinction is made between the enthalpy of stored refrigerant and the flowing refrigerant, and convergence is defined by the prediction of the same enthalpy state at each node by two forms of the flow-stream enthalpy. This approach required the assumption of a uniform refrigerant liquid level at the start of the transient operation. No comments were made about the pressure drop in the evaporator.

Application of the model:

The model developed is applicable to any vapor compression system, with a hermetic compressor, refrigerant-in-tube heat exchangers and orifice type or thermostatic expansion devices. The model cannot be used for shell and tube heat exchangers, dynamic compressors or capillary tubes. The model is refrigerant independent, with suitable changes required in the thermodynamic property relations for the specific refrigerant used. Also, the model imposes no constraint on the secondary fluid of the heat exchanger.

Discussion:

Validation:

The model developed was verified for internal consistency and detailed validation against specific test set-ups is reported in MacArthur and Grald [1987]. Start-up, steady-state and shutdown predictions are presented in the [1987] work and shown to match the measurements well.

Comments:

The development from first principles was found to be well presented and transparent. The absence of the momentum principle in the analysis of the heat exchangers appears to have simplified the model by not accounting for pressure drops in the heat exchangers.

References:

Benton R., MacArthur J.W., Mahesh J.K. & Cockroft J.P., 1982, "Generalized modeling and simulation software tools for building systems", ASHRAE Transactions, Vol. 88, Part 2, pp. 839-856.

Dhar M. & Soedel W., 1979, "Transient analysis of a vapor compression refrigeration system", XV International Congress of Refrigeration, Venice.

MacArthur J.W., Benton R. & Mahesh J.K., 1981, "Generalized engineering modeling and simulation tool – GEMS", Honeywell Final Report, Vol. I, II, III.

MacArthur J.W., Meixel G.D. & Shen L.C., 1983, "Application of numerical methods for predicting energy transport in earth contact system", Journal of Applied Energy, Vol. 13.

MacArthur J.W. & Grald E.W., 1987, "Prediction of cyclic heat pump performance with a fully distributed model and a comparison with experimental data", ASHRAE Transactions, Vol. 93, Part 2, pp. 1159-1178.

5.1.6 Cyclic Characteristics Of A Typical Residential Air-Conditioner: Modeling Of Start-Up Transients

W.E. Murphy & V.W. Goldschmidt, 1985
ASHRAE Transactions, Vol. 91, Part 2A, pp. 427-444

Equipment Description:

Only the compressor, condenser, expansion device and liquid line of a vapor compression refrigeration system were modeled. The compressor is hermetic, the expansion device is a set of 4 parallel capillary tubes and the condenser is air-cooled.

Purpose

This model was developed to study alternate designs of condenser and liquid line for their impact on start-up performance.

Assumptions:

- Uniform heat transfer coefficients in the superheated, two-phase and sub-cooled regions of the condenser
- Neglects the initial period when the condenser is filled with superheated vapor.
- A fixed (3 sec) time for condenser outlet to become sub-cooled.
- Complete 2-phase region in the condenser is characterized by a single constant quality, irrespective of the size of the 2-phase region.

Mathematical Description:

The capillary tube is modeled in the classical way using the pressure drop and Fanno flow relations to determine the mass flow rate. The model also considers the possibility of the existence of an initial sub-cooled length in the capillary tube by using a simple in-tube pressure drop equation.

The compressor is modeled using steady-state equations, defining power as a function of the suction pressure and pressure ratio only. The mass flow rate is determined as a function of the suction density and pressure ratio. These correlations are obtained by incorporating friction and other losses into the compressor's ideal power relation and obtaining an appropriate curve-fit. Further detail of the compressor model is not provided.

The condenser is modeled assuming saturated vapor, two-phase or liquid exiting. The condition at the exit is obtained from the liquid line model and a mass balance between the liquid line and capillary tubes. Sub-cooling is considered to start in the condenser when the liquid line is flooded with liquid. The condenser is treated in three zones, i.e. superheated, two-phase and sub-cooled. The heat transfer coefficients are obtained from standard empirical correlations. The two-phase pressure drop is determined over discrete-quality portions of the two-phase region. In the single phase regions, the pressure drop is calculated using the standard Darcy friction factors. The air-side heat transfer is determined by an energy balance through the condenser wall using the conductive resistance and capacitance of the condenser wall and the convective resistance of the air.

Solution Technique:

The solution consists of determining the future states – namely the condenser pressure, the evaporator pressure, the mass flow rate, the condenser tube wall temperature and compressor work from the existing states. Assuming an initial condition with the system off, the first 0.4 seconds during which the condenser pressure reaches the saturation pressure are ignored. With saturated vapor exiting the condenser, the liquid line model is used to determine the back-up of liquid into the condenser. Until then, the condenser continues to operate only in the de-superheating and two-phase modes, the capillary tube operates only with two-phase vapor and the compressor suction conditions are determined from the evaporator data (which remains external to the present model).

Once the liquid line is completely filled with liquid, the capillary tube model is used to determine the mass flow rate and exit pressure. The condenser is treated first for the superheated and then the sub-cooled regions. These regions are discretized to obtain temperature profiles. Details of this discretization are not included. The two-phase region is estimated by removing the de-superheating and sub-cooled regions from the total condenser volume. The mass balance in the liquid line gives the change in the sub-cooled region.

Application of the model:

This model applies exclusively to an existing air-conditioner and is not a general purpose model. This is particularly so for the condenser model which makes a re-arrangement of the tube-passes of the original condenser. Since the evaporator is included from external data, no refrigerant inventory is done. Also, the expansion device model is for a capillary tube and is modeled for a set of 4-tubes in parallel. As such, this model cannot be extended to a different system. However, the principles developed could be used to model larger similarly configured systems.

Discussion:

Validation:

The model developed was compared to the air-conditioner and found to exhibit the same trends as measured. However, during the first 1 minute of operation, significant differences were found in the refrigerant pressures. This was explained as different cycles being used between the measured and computed data, but the explanations were not satisfactory.

Comments:

The presentation of the solution is not very clear. Most of the model appears to be based on empirical correlations and curve-fits and inadequate information is provided about the type or accuracy of the fits. This model was developed more to study the one specific air-conditioner that was used as the basis and not as a general vapor compression refrigeration system.

References:

Murphy W.E. & Goldschmidt V.W., 1984, "Transient response of air conditioners – a qualitative interpretation through a sample case", ASHRAE Transactions, Vol. 90, Part 1A.

Goldschmidt V.W. & Murphy W.E., 1979, "Transient performance of air conditioners", Proc. of Technical Groups, New Zealand Institute of Engineers, Vol. 5 (Issue 4, ISSN-0111-1358), pp. 715-738.

Traviss D.P., Rohsenow W.M. & Baron A.B., 1973, "Forced convection condensation inside tubes: A heat transfer equation for condenser design", ASHRAE Transactions, Vol. 79, Part 1, pp. 157-165

Josiasson N.J., 1978, "Simulation of condition sequence during start-up of an evaporation refrigerating system." Proc. Purdue Compressor Technology Conference, pp. 309-316
Geary D.F., 1975, "Return bend pressure drop in refrigeration systems", ASHRAE Transactions, Vol. 81, Part 1, pp.250-256.

5.1.7 Cycling Characteristics Of A Residential Air Conditioner – Modeling Of Shutdown Transients

Murphy W.E. & Goldschmidt V.W., 1986
ASHRAE Transactions, Vol. 92, Part 1A, 186-202

Equipment Description:

The equipment modeled here are the heat exchangers, the expansion device (capillary tubes), a pressure equalization valve and the plumbing. The focus of this work is on the dynamics of the refrigerant at shutdown. The compressor is not modeled.

Purpose

This model is built to study cyclic losses by focusing on the shutdown transients.

Assumptions:

- Plumbing is adiabatic
- Flow through the compressor is negligible
- Air-side heat transfer is by natural convection.
- Heat exchanger walls are in thermal equilibrium with the refrigerant.
- Back flow through the pressure equalization valve is saturated vapor.

Mathematical Description:

In this model, both heat exchangers are treated as tanks connected by two parallel flow paths, namely the liquid line containing the capillary tubes and another line containing the pressure equalization valve (PEV). Refrigerant migration through the compressor is considered negligible and hence the suction line is merely lumped with the evaporator. The liquid line is treated as an additional component, discretized along its length, and with boundary conditions corresponding to the condenser exit and the capillary tube inlet conditions. In the liquid line, the re-distribution of enthalpy is modeled as a simple wave traveling from the condenser to the capillary tubes, in that the enthalpy at any node at a particular time-step is the same as the enthalpy at the preceding node at the earlier time-step. The mass distribution through the liquid line is modeled separately for liquid flow and for two-phase flow, with the switchover being triggered by the end of liquid out-flow from the condenser.

Both the heat exchangers are modeled by mass and energy balances, in time-discretized form, with heat transfer to/from the ambient.

The PEV is modeled as an orifice, and all pressure drops in the condenser, discharge line, compressor shell, the PEV and the suction line are lumped together with the overall pressure difference between the condenser and evaporator pressures in determining the mass flow rate through the PEV. Also, the refrigerant flowing through the PEV is considered to be in saturated vapor form at all times.

Solution Technique:

The solution proceeds sequentially by first fixing the initial conditions of the system at shutdown. The flow rates through the capillary tube and the PEV are then determined. Based on the loss of mass and energy during the time step, new conditions in the condenser are obtained. The distribution of mass and enthalpy along the liquid line are updated. Based on the gain of mass and energy into the evaporator, new conditions in the evaporator are determined. This sequence is continued until the pressures in the evaporator and the condenser are equal. The mass and energy balances in both the heat exchangers are integrated by the implicit Euler method.

Application of the model:

The model developed focuses on the phenomenon of shutdown rather than on the specifics of any particular equipment. Since the models of the heat exchangers are simply ‘tanks’ they could be used as-is for literally any construction. However, the use of a capillary tube for the expansion device restricts the model to this. With other active expansion devices, the behavior of the system would be significantly altered. Also, the model considers the presence of a PEV which provides an additional path for refrigerant migration. In systems where this is not present, the approach to steady state would be delayed appreciably.

Discussion:

Validation:

The model developed was validated against an existing air-conditioner set-up for which shutdown data was collected earlier along with start-up data (Murphy et al [1985]). With the exception of the first 5 seconds, the predictions of the model compared well with the measured data. The discrepancies during the first 5 seconds are explained as due to the rapid equalization of the pressures (by elimination of the pressure drops) within the heat exchangers immediately after the shutdown. Having validated the model, the shutdown behavior was also studied with different refrigerant charges, different openings of the PEV and thermally bulkier heat exchangers.

Comments:

The development of the model is very lucid and the phenomena occurring during shutdown are presented exhaustively.

References:

Erth R.A., 1969, "Two-phase flow in refrigeration capillary tubes: analysis and prediction", PhD Thesis, Purdue University, West Lafayette, IN.
Goldschmidt V & Murphy W., 1979, "Transient performance of air conditioners", Proc. of Technical Groups, New Zealand Institution of Engineers, Vol. 5.
Murphy W. & Goldschmidt V., 1984, "Transient response of air-conditioners – a qualitative interpretation through a sample case", ASHRAE Transactions, Vol. 90, Part 1.
Murphy W. & Goldschmidt V., 1985, "Cyclic characteristics of a typical residential air-conditioner – modeling of start-up transients", ASHRAE Transactions, Vol. 91, Part 2.

5.1.8 Prediction Of The Transient Response Of Heat Pumps

S.M. Sami, T.N. Duong, Y. Mercadier, N. Galanis, 1987
ASHRAE Transactions Vol. 93, Part 2, pp. 471-490

Equipment description:

The equipment modeled here is a vapor compression refrigeration system using either R12 or R22 as the refrigerant. The components modeled are a shell and tube and an air-cooled condenser, thermostatic expansion valve and a capillary tube, a direct expansion, a shell and tube and a coil type evaporator and a reciprocating compressor.

Purpose

This model was constructed to meet the requirement of a model that could capture frost formation, oil migration, flow reversals, flashing and compressor hunting.

Assumptions:

- Refrigerant flow in the compressor is compressible and the processes are isentropic or polytropic.
- Refrigerant vapor behaves as an ideal gas in the compressor
- Kinetic and potential energies of refrigerant during compressor strokes are negligible
- Liquid and vapor in the heat exchangers are in thermal equilibrium

Mathematical description:

The total system is divided into a number of lumped-parameter control volumes. The paper describes the details of the refrigerant control volumes and applies the mass/energy conservation equations, and where applicable, the momentum equation. The condenser is modeled as a combination of the three different phase regimes namely, the fully superheated, the two-phase and the sub-cooled. The vapor and liquid phases in each regime are modeled by a set of coupled mass and energy balances. The same approach is followed in the evaporator. The correlations for internal and external heat transfer coefficients, thermo-physical and thermodynamic properties of refrigerants, and equations for external (non-refrigerant side) heat transfer are modeled from references.

Also, phenomena such as oil-migration, water and frost properties over the evaporator surface, void fraction and slip, pressure drop have been modeled from references and the details have not been described in this work.

Solution technique:

Sets of coupled 1^{st} order linear differential equations are obtained for each control volume from the applicable conservation equations, which are then solved (not described how) to obtain the response of that control volume. The characteristics of the components were coded in FORTRAN IV modules. No detail is presented on the numerical methods used in the solution, except that the total system is simulated by time-wise integration of the constituent models, starting with the condenser and an initial guess of the pressure, temperature and flow rate of the refrigerant.

Application of the model:

By selection of the appropriate modules and an event-trapping algorithm, the set of models developed in this work can be used to model any vapor compression refrigeration system made up of a combination of components identified above under *Equipment Description* and is also limited to such systems. This cannot be used to model systems with centrifugal compressors, or with systems not operating on R12 or R22.

Discussion:

<u>Validation:</u>

The validity of this model has been established by comparison with experimental results of an R12 based system consisting of a single-cylinder reciprocating compressor, a shell and tube condenser, an evaporator and a throttling valve, with capacity from 1kW-5kW, tested by Yasuda et al (1982). Validation has also been done against an air-water heat pump system experimental study by Hamel (1981) of start-up transients. In both cases, the results from the simulation compare well with the experimental data.

Comments:

The work presented is comprehensive in the phenomena and conditions that it can handle. However, details are missing about how the different flow regimes have been modeled. The focus of the work is the development of equations for the refrigerant-side interactions alone and its interaction with the rest of the system is not adequately clarified. Also no details are presented about the numerical solution of the model. Some typographical errors in the references and in the equations were found. Referencing is not clear.

References:

Yasuda Y. Touber S and Machielson, 1982, "Simulation model of a vapor compression refrigeration system", ASHRAE Transactions Paper No. 2787, pp. 408-425

Dhar M., 1978, "Transient analysis of refrigeration system", PhD Thesis, Purdue University

Sami S.M., Duong T., 1986, "Development of DAHP; transient computer program to simulate heat pump performance", T.R. MEC/86/2

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5.1.9 Performance Of Engine-Driven Heat Pumps Under Cycling Conditions

R.W. Rasmussen, J.W. MacArthur, E.W. Grald and G.A. Nowakowski, 1987
ASHRAE Transactions, Vol. 93, Part 2, pp. 1078-1090

Equipment Description:

The equipment modeled here consists of a natural-gas fueled engine as the prime mover, an open reciprocating compressor, the evaporator, condenser and engine radiator, accumulator, expansion device and control devices for the system. The focus of this model is the effect of cycling rate on system efficiency.

Purpose

This model was developed to propose an IC engine as an alternate prime mover for the vapor compression system.

Assumptions:

- Engine performance modeled at constant air-fuel ratio
- Engine operating at best spark timing
- Assumptions inherent in the components of the refrigeration system are not explicitly mentioned.

Model Description:

Since the intent of this work was to study the effect of cycling rate on vapor compression system behavior, mathematical model descriptions are not detailed. The model of the engine is developed based on dynamic behavior of two natures, namely inertial dynamics caused by speed changes and thermal dynamics caused by load changes. The engine model is represented as a map of the brake specific fuel consumption and the shaft power as functions of the speed and load. The components of the heat pump are imported from an earlier (1986) work by the 2nd and 3rd authors, but the citation is missing from the references. It is expected that the system models were on the lines of MacArthur and Grald [1987] (reviewed elsewhere in this report). The heat exchanger models were sized to specific cooling capacity conditions. The compressor model was imported from manufacturer's data for a variable-speed unit.

Solution Technique:

Although no details have been presented explicitly about the solution technique, it is expected that it is the same as in MacArthur and Grald [1987]. The simulation consisted of running the model from an ambient-equilibrium condition through successive repetitions of on/off cycling under different conditions for each cycle, each simulation being run at one set of the above two parameters, and the engine reaching a constant steady state speed. The simulations were run in cooling and heating mode and the cycling performance was studied. The system performance was characterized by a part load factor (PLF), defined as the ratio of the cyclic COP to the steady state COP, the cyclic COP being defined as the ratio of useful output during the cycle to the *total* energy input, inclusive of fans & pumps

Application of the model:

The work presented appears applicable to any combination of vapor compression system driven by an IC engine. Alternate models would (maybe) be required for the components of the vapor compression system, when modeling a different configuration. The approach, however, is specific in that specific maps were used for the engine performance, and generic in that alternate maps could be developed for different engines/operating conditions.

Discussion:

<u>Validation:</u>

The models developed were validated against existing laboratory equipment instrumented and tested for the purpose. The details of the validation are also referenced to MacArthur and Grald [1987]. The complete system was simulated for different sets of conditions of two parameters, namely:

- Indoor/outdoor temperature difference and
- Nominal cycling rate

The components were validated against laboratory set-ups before putting them together as a system. The significant conclusions drawn are that:

- the effect of cycling rate becomes less important as temperature difference between indoor/outdoor air becomes small
- in the heating mode with the fixed orifice, there exists an outdoor temperature (for a given indoor temperature) below and above which the PLF increases.

Comments:

The description of the heat pump response to on/off cycling is presented exhaustively. The correlation of the existence of a 'critical' outdoor temperature to expected physical phenomena of refrigerant migration is interesting.

References:

Jennings M.J. & Blumberg P.M., 1986, "A dynamic model of a spark-ignited natural gas fueled heat pump engine" Unpublished ITI Report No. 86-105, June 16.

MacArthur J.W. & Grald E.W., 1987, "Prediction of cyclic heat pump performance with a fully distributed model and a comparison with experimental data", ASHRAE Transactions Vol. 93, Part 2.

5.1.10 Prediction Of Cyclic Heat Pump Performance With A Fully Distributed Model And A Comparison With Experimental Data

J.W. MacArthur & E.W. Grald, 1987

ASHRAE Transactions Vol. 93, Part2, pp. 1159-1178

Equipment Description:

The equipment modeled here are the heat exchangers and the accumulator. The modeling focuses on the refrigerant side primarily. The secondary fluid is not specified and representative values for it and the tube heat transfer are assumed.

Purpose

This model was developed as a research tool to aid in the evaluation of system performance.

Assumptions:

- Flow through the tubes is 1-D and homogenous
- Expansion is isenthalpic
- Compression is isentropic
- Axial heat conduction in the refrigerant is negligible
- Effects of pressure wave dynamics are negligible
- The work due to pressure change in the refrigerant is negligible

Mathematical Description:

General equations of mass, momentum and energy conservation are reduced to simpler forms by applying the above assumptions. The momentum equation is merged with the energy equation by simple algebra and a ‘modified’ form of the energy equation is developed, which eliminates the necessity of working explicitly with the momentum equation. These two (mass and energy) equations are discretized form. Equations for the energy balance of the heat exchanger wall and the secondary fluid are similarly discretized. Ultimately, for each small control volume in the heat exchangers there are 4 coupled finite difference equations relating refrigerant mass flow, enthalpy, heat exchanger wall temperature and secondary fluid outlet temperature at any time with their respective values at the previous time instant and to the previous control volume. The accumulator is treated as a single additional control volume at the end of the evaporator. The models of the expansion valve and the compressor are not included in this work. These have been imported from references.

Solution Technique:

For solving the above 4 equations for each element, an initial distribution of refrigerant is assumed or initialized in the heat exchanger. The boundary conditions for the condenser are set as the compressor discharge enthalpy and mass flow rate and expansion valve mass flow rate, while for the evaporator, they are set as the expansion valve mass flow rate and enthalpy and the mass flow rate into the accumulator.

At each time step, the saturated refrigerant properties for the current pressure and heat transfer coefficients are calculated. The mass and energy balance equations for the refrigerant are solved alternately for each control volume, along the length of the heat exchanger to get a mass-flow field and an enthalpy field. The heat exchanger wall and secondary fluid temperatures are also determined. This process is repeated until convergence of the enthalpy field. The calculated mass and enthalpy fields are then compared with the boundary conditions and the pressure change to reduce the difference is determined (by a secant method). The sequence of re-calculating the exit conditions is repeated with the updated pressure until the boundary conditions match. The total values are then obtained by integrating over all the control volumes. This process is repeated through time.

Application of the model:

The models developed here are applicable to counter flow and parallel flow heat exchangers, with no restriction imposed on the secondary fluid. In case of cross-flow heat exchangers, a modification is required in the finite-difference equations. However, the same algorithm could be used.

This model applies only to refrigerant-in-tube heat exchangers and cannot be used for shell & tube constructions or other configurations where the refrigerant condenses/boils on the outside of the tube.

Discussion:

Validation:

This model has been validated against four system configurations:

- a 3-ton water-to-water heat pump with a hermetic compressor
- a 2-ton air-to-air heat pump with an open drive compressor
- a 3-ton air-to-air heat pump with an open drive compressor
- a 3-ton air-to-air heat pump with an IC engine drive.

In all cases, the predictions match the observations.

Comments:

The work presented is very clear in the mathematical reduction from the generalized form to the discretized form and makes re-construction of the mathematical model by the reader a possibility. However, it was felt that the final forms of the imported compressor and expansion device models ought to have been presented, for completeness.

References:

Rasmussen R.W., Grald E.W., MacArthur J.W. & Ruohonieni T.J., 1986 "Gas Heat Pump System and Component Efficiency and Reliability Improvement Development: Controls & Valves", Final Report to Gas Research Institute.

Dhar M. & Soedel W., 1979, "Transient Analysis of a Vapor Compression Refrigeration System.", XV International Congress of Refrigeration, Venice.

Chi J. & Didion D., 1982, "A Simulation of the Transient Performance of a Heat Pump", International Journal of Refrigeration, Vol. 5, No. 3.

Brasz J.J., & Koenig K., 1983, "Numerical Methods for the Transient Behavior of Two-Phase Flow Heat Transfer in Evaporators and Condensers", Numerical Properties and Methodologies in Heat Transfer: Proceedings of the Second National Symposium, pp. 461-476.

MacArthur J.W., 1984, "Analytical Representation of the Transient Energy Interactions in Vapor Compression Heat Pumps.", ASHRAE Paper, HT-84-19, Vol. 90, Part 1.

5.1.11 Simulation In Transient Regime Of A Heat Pump With Closed-Loop And On-Off Control

Vargas J.V.C. & Parise J.A.R., 1995
International Journal of Refrigeration, Vol. 18, No. 4, pp. 235-243

Equipment Description:

The equipment modeled here is a simple vapor compression air-conditioning system, (including the conditioned space), with a reciprocating compressor driven by a DC servomotor. The DC servomotor (driving the compressor), the temperature sensor and an alternate control action are also modeled.

Purpose

This model was developed to compare alternate control methods that would yield lower energy consumption.

Assumptions:

- Conditioned space air-mass is constant and conditions are uniform
- Pressure drops in the heat exchangers are negligible
- Off-cycle refrigerant migration is instantaneous
- Heat exchanger wall resistance and capacitance are neglected.
- Compression is polytropic
- Heat exchange through cylinder walls is negligible
- Expansion is isenthalpic
- Superheated refrigerant behaves as an ideal gas.

Mathematical Description:

All the thermal components of the system are modeled simplistically in lumped parameter form. The system is divided into 7 control volumes, namely, the conditioned space, air-side of evaporator, refrigerant side of evaporator, compressor, expansion valve, refrigerant side of condenser and air-side of the condenser. Each of these control volumes is characterized by space-averaged state variables as functions of time only. Combined mass and energy balances are applied to each of these to obtain a set of first order linear differential equations in time. Within the heat exchangers, in the two-phase zones, a constant quality is determined from a space and time average, assuming a linear variation of quality over the heat exchanger length. The heat transfer across the heat exchangers is approximated as occurring between the average refrigerant temperature and the mean of the inlet and outlet air temperatures, with a global heat transfer conductance. Within the conditioned space, the heat loads are distinguished as heat gains through the walls and equipment load.

The compressor is modeled by the volumetric and polytropic efficiency method. No comment was found regarding determination of the polytropic efficiency. The mass flow rate and power consumption of the compressor are determined. Capacity control is achieved by speed variation of the driving motor

The thermostatic expansion valve is modeled from its flow area as a linear function of the instantaneous superheat.

The DC servomotor is modeled electrically as an RL circuit and mechanically by the angular momentum equation. This model couples to the compressor through the power consumption term.

The closed-loop control action is based on a power law, instead of the traditional PID, with the idea that it is simpler and more efficient. This power-law control-action is used to vary the motor speed, based on the error signal of conditioned space actual and desired temperatures.

Solution Technique:

The model developed above was solved using the Runge-Kutta Fehlberg fourth-fifth order method with controlled step size.

Application of the model:

The significant part of the model developed here is the power-law control-action, which appears applicable to any refrigeration system. This being the focus, the model for the rest of the system is highly simplified. However, the simplification of the model is not a constraint on the applicability of the control method developed.

Discussion:

The model was run with fixed and variable thermal load in the conditioned space. No validation experiments are reported. The simulations were run using the power-law control action developed in this work, and a traditional on-off cycling based on conditioned space temperature. It is concluded that the use of the power law control action generates a savings of about 11% in the system's energy consumption by avoiding the oscillations in the system inherent with the on-off control method.

References:

Chi J. & Didion D.A., 1982, "A simulation model of the transient performance of a heat pump", International Journal of Refrigeration, Vol. 5, No. 3, pp. 176-184.

MacArthur J.W., 1984, "Transient heat pump behavior: a theoretical investigation", International Journal of Refrigeration, Vol. 7, No. 2, pp. 123-132.

Eastop T.D. & McConkey A.M., 1978, "Applied Thermodynamics for Engineering Technologists" 3rd Edition, Longman (Chap. 17)

Neale D.F., Wang Y.T., Wilson D.R. Green R.K. & Searle M., 1981, "Microprocessor based control system for heat pumps", Third International Conference on Future Energy Concepts, London, pp. 238-241.

Kuo B.C., 1987, "Automatic Control Systems", Prentice Hall International Editions (Chap. 4)

5.1.12 Improvements In The Modeling And Simulation Of Refrigeration Systems: Aerospace Tools Applied To A Domestic Refrigerator

L. Ploug-Sørensen, J.P. Fredsted & M. Williatzen, 1997

International Journal of HVAC&R Research, Vol. 3, No. 4, pp. 387-403

Equipment Description:

This paper presents a model of a domestic refrigerator developed using the SINDA/FLUINT (Systems Improved Numerical Differencing Analyzer with Fluid Integrator) software, with the focus on applicability and features of SINDA/FLUINT which is an established simulation tool in the aerospace industry. The domestic refrigerator modeled consisted of a hermetic compressor, an evaporator with a cabinet, a capillary tube attached to the suction line and an air-cooled condenser.

Purpose

This model was developed to demonstrate the capabilities of the software SINDA/FLUINT and propose it as a model development environment.

Assumptions:

The assumptions applied in the refrigerator model are:

- Spatial temperature variations in the conditioned space negligible
- Oil in the refrigerant neglected, i.e. the refrigerant is pure
- Only dry analysis of the air is done
- Frosting and thawing of the evaporator not considered
- Load variations and door openings not considered
- Constant volumetric efficiency
- Constant isentropic efficiency
- Flow through evaporator is homogenous
- Flow through condenser is slip-flow

The assumptions inherent in SINDA/FLUINT are:

- Thermodynamic equilibrium exists in each lump
- Boiling is either film or nucleate

- Condensation is as described by Rohsenow
- Darcy friction factor for single phase pressure drop is from Churchill's equation
- Work associated with pressure changes in the refrigerant is negligible
- Viscous dissipation is negligible

Mathematical Description:

In SINDA/FLUINT, the model is developed in two directions – the thermal and the fluid. For each model, the system is divided into 'transporters' and 'containers', with the user choosing the type of transporters and/or containers required for any specific system, depending upon the inclusion/exclusion of inertial terms. The containers are modeled using the conservation of mass and energy, while the transporters are modeled using the conservation of momentum. These are applied similarly to the thermal and the fluid models. SINDA/FLUINT also has built-in functions for single phase and two-phase heat transfer coefficients and pressure drops.

A domestic refrigerator model is constructed in SINDA/FLUINT's graphical user interface. The compressor model consists of several lumps, namely suction side volume of gas heated in the hermetic shell, suction side volume of gas entering the compressor directly, discharge side volume of gas in the stroke volume, a thermal model based on constant (different for different compressors and refrigerants) of the isentropic and volumetric efficiencies. These constants were determined by correlations against compressor speed and condensing/evaporating temperatures; the data for these correlations are referenced to Rasmussen [1997]. These lumps are connected by fluid-flow and heat-flow connectors. The compressor work is calculated using the volumetric-efficiency formulation.

The evaporator model also consists of several lumps, with only one lump directly connected to the rest of the system. This lump has only mass conservation and no mass/energy/momentum storage. This lump connects to the discretized lumps of the evaporator tube, which are all connected sequentially with connector lumps. Each of the tube lumps are evaluated using mass and energy conservation and the connectors are evaluated with mass and momentum conservation. The energy balances for all the lumps are connected to another lump representing the cabinet space which evaluates the complete heat transfer from the evaporator. The cabinet space is modeled for energy conservation only and connects to a last lump which represents heat transfer to the ambient from the cabinet space. This last lump is a boundary node having infinite thermal capacitance.

The condenser model is similar to the evaporator model with the exception that the cabinet space lump is missing and all the discretized lumps connect (thermally only) directly to the boundary ambient lump. Also, the discretization of the condenser tube is done as a 'parallel' tube, with one tube carrying only vapor and one carrying only liquid refrigerant. These parallel sets of lumps are presumed to be internally connected (fluid only) one-to-one.

The capillary tube is modeled as a very small evaporator, exchanging heat with the suction line only.

The suction line is also modeled on the lines of the evaporator, except that the heat exchange occurs with the capillary tube over a part of its length and also with the ambient over its entire length.

Solution Technique:

Although the solution technique used by SINDA/FLUINT is not detailed, it appears that the solution marches in time through the components and at each time step the sets of equations of all the lumps of each component are solved simultaneously to convergence. It is not clear whether an explicit or implicit method is employed in the finite difference formulation or how (and if) any optimization is done for the time step.

Application of the model:

The model developed here demonstrates the capabilities of SINDA/FLUINT as a modeling tool for standard thermal equipment. The model is constructed from a set of general purpose elements which are based on the fundamental laws of mass/energy/momentum conservation. As such, they can be combined in different ways to build a wide variety of thermal equipment. It is reported that the execution of the simulation progressed close to real time in the ON period, but caused severe numerical problems during OFF periods when the mass-flows came close to zero. This necessitated overall simulation times longer than real time.

Discussion:

The model developed was validated against a 325L refrigerator running on R600a, with a 2L hermetic compressor running at a constant speed of 450rpm. The model simulation was compared against the system response for compressor start-up, cycling based on cabinet space temperature and compressor shut-down. The simulation showed fairly good correlation with measurements from the same refrigerator. Fine-tuning was achieved by adjusting the outside-of-tube heat-transfer coefficients of the evaporator and condenser and that between the cabinet and the ambient. This adjustment is justified by the difficulty in accurate estimation of the heat transfer areas.

References:

Cullimore B., Ring S.G., Goble R.G. and Jensen C.L., 1996, "Sinda/Fluint Users Manual for Version 3.2", Cullimore and Ring Technologies, Inc., Littleton, CO.

Dossat R.J., 1991, "Principles of Refrigeration" 3rd Ed. (Chap. 12) Prentice Hall, Englewood Cliffs, NJ.

Rasmussen B.D., 1997, "Variable Speed Hermetic Reciprocating Compressors for Domestic Refrigeration", PhD Thesis No. ET-PhD. 97-03, Department of Electrical Engineering, Technical University of Denmark, Copenhagen.

Rohsenow W.M. and Hartnett J.P., 1973, "Handbook of Heat Transfer", McGraw-Hill, New York.

5.1.13 Development And Validation Of A Real-Time Transient Model For Air-Conditioners With On/Off Control

Todd Rossi & James E. Braun, 1999

Proc. 20th International Congress of Refrigeration (Sydney), Paper no. 743

Equipment Description:

The equipment modeled here is a rooftop air-conditioning unit operating on the vapor compression cycle and consisting of an air-cooled condenser, a direct expansion evaporator, a single-stage hermetic compressor, a fixed orifice or thermostatic expansion valve, an accumulator, a receiver and the plumbing.

Purpose

This model was developed to meet the need for a dynamic system model that could run faster than real-time, while capturing all significant transients.

Assumptions:

- Refrigerant flow is 1D and homogenous.
- Refrigerant pressure drops in the heat exchangers are neglected.
- Heat exchanger material has negligible resistance and finite capacitance.
- The heat exchanger is a single finned-tube in cross flow with air.
- Uniform heat transfer coefficients on refrigerant side.
- Wet-air Lewis number is unity.

Mathematical Description:

The heat exchangers are modeled on a fixed-boundary finite difference formulation. Elemental mass and energy balances are written for each element of the heat exchanger, including the heat transfer equations between the refrigerant, the tube-wall and the air. The momentum equation does not appear as the pressure drops are considered negligible. The heat exchanger model uses the inlet and outlet refrigerant flow rates and the inlet enthalpy as boundary conditions.

The discretization of each heat exchanger is done uniformly across the refrigerant, heat exchanger wall and the air. The heat exchanger material energy balance couples with the refrigerant energy balance through the wall temperature.

The air-side heat transfer is modeled differently for heat transfer (dry analysis) and heat and mass transfer (wet analysis). The former is based on the temperature potential between the wall and the air, while the latter is based on the enthalpy potential between the refrigerant and the air. Both situations are modeled using an effectiveness-NTU formulation. When using the heat exchanger model for the condenser, only the dry analysis is done. For the evaporator, a selection of wet or dry analysis is made depending on the temperature of the heat exchanger wall being above or below the dew point temperature of the incoming air.

The instantaneous conductances on the air-side and the refrigerant-side are obtained from a correlation determined from a regression analysis of available data. The air-side conductance also incorporates the fin efficiency.

The compressor model is built from compressor data using polynomial fits. An energy balance is applied between the refrigerant entering the compressor shell, the compressor shell itself and the heat transfer to the ambient to determine the shell losses.

The expansion devices are modeled from first principles. The orifice is modeled without dynamics while the thermostatic expansion valve considers the transient response of the sensing bulb. The bulb's dynamic response is modeled as a simple thermal-capacitance. The dynamics of the diaphragm are neglected.

The receiver, accumulator and the plumbing are modeled as single node heat exchangers attached to their preceding or following components.

Solution Technique:

The elemental mass and energy balances are algebraically manipulated to obtain a set of algebraic differential equations in exact time-derivatives of the elements pressure and enthalpy. These equations are solved simultaneously, at any time step, to obtain the time derivatives of the state variables. Similar state-variable derivatives are obtained for all the components. These derivatives are then integrated using the 4th order Runge-Kutta method to obtain the input states for the next time step. This process is progressed till the end of the transient simulation.

To decrease the simulation time, the integration time-step is dynamically adjusted by an external algorithm. The time step begins at a small value at start-up or shutdown and increases in a negative exponential trend towards a steady state value. The rate of change is different for the cases of the system moving from on to off and from off to on.

Application of the model:

The model developed for the heat exchangers is a fairly general one and applicable to any kind of refrigerant-in-tube heat exchanger. With a few changes, the approach may also be applied to shell-and-tube heat exchangers. Since the compressor is modeled from a map, the present model is restricted to the system described under *equipment description*. The orifice and the thermostatic expansion valve, which are among the most common of expansion devices are built from basics. No imposition is made on the refrigerant or the secondary fluid.

Discussion:

Validation:

The model developed was validated against an existing 3-ton rooftop air-conditioner from which exhaustive test data had been collected. Fine-tuning of the model was done by adjusting the heat-transfer coefficients from steady-state data. The validation was done against measured data for the case of the system switching ON after an extended period of being OFF, akin to start-up. The prediction was found to compare well with the measured data.

Comments:

The heat exchanger model developed is flexible yet simple as it replaces the partial derivatives with total derivatives. Also, the dynamic calculation of the integration time step is done explicitly, which reduces the dependence on the solver and saves valuable simulation time.

References:

Braun J.E., Klein S.A. & Mitchell J.W., 1989, "Effectiveness models for cooling towers and cooling coils", ASHRAE Transactions, Vol. 95, Part 2, pp. 164-174.
Breuker M.S., 1997, "Evaluation of a statistical, rule-based fault detection and diagnostics method for vapor compression air-conditioners", MS Thesis, School of Mechanical Engineering, Purdue University.
Breuker M.S, & Braun J.E., 1998, "Common faults and their impacts for rooftop air-conditioners", International Journal of HVAC&R Research, Vol. 4, NO. 3, pp. 303-318.
Zivi S.M., 1964, Estimation of steady-state steam void fraction by means of the principle of minimum entropy production", Journal of Heat Transfer, No. 86, pp. 247-252.
MacArthur J.W., 1984, "Analytical representation of the transient energy interactions in vapor compression heat pumps", ASHRAE Transactions, Vol. 90, Part 1.
MacArthur J.W., 1984, "Theoretical analysis of the dynamic interactions of vapor compression heat pumps", Energy Conservation Management, Vol. 24, No. 1, pp. 49-66.

5.1.14 Non-Steady-State Modeling Of A Water-To-Water Heat Pump Unit

Morten Christian Svensson, 1999

Proc. 20th International Congress of Refrigeration (Sydney), Paper No. 263

Equipment Description:

The equipment modeled here is a complete vapor compression system consisting of a shell and tube condenser, a shell and tube evaporator, an electrical motor driven reciprocating compressor and a thermostatic expansion valve.

The focus of this work is on the response of the system to changes in the operating conditions.

Purpose

This model was developed for use as an on-line observer for control systems.

Assumptions:

The modeling assumptions are:

- The total volume of vapor in the condenser is constant
- Saturated refrigerant enthalpies are independent of pressure
- Longitudinal conduction in the heat exchanger pipes is negligible
- Uniform heat transfer coefficients in each phase zone in the condenser
- Uniform heat transfer coefficient in all of the evaporator
- Constant superheat
- Thermal mass of the compressor is negligible
- Spatial pressure variations are negligible

Mathematical Description:

The condenser is modeled as consisting of two zones: the condensing and the sub-cooling, both of which are assumed to have fixed volumes. Energy balances and mass balances are obtained for each of these zones as a set of coupled ordinary differential equations. These are then combined to give an equation in the pressure derivative in terms of the tube-wall temperature, the incoming mass flow rate and saturated condition refrigerant properties. The tube-wall and the water volume are discretized into ‘n’ volumes in the condensing zone and 1 volume in the sub-cooling zone and finite difference forms of the energy balance equations are used, and for each of these ‘n’ volumes, the refrigerant side conditions are considered identical. A similar approach is adopted for the evaporator, except that the complete evaporator is modeled as a single zone. The expansion device is modeled as a perfect isenthalpic device maintaining constant superheat. The compressor is modeled in a standard isentropic efficiency form.

Solution Technique:

The details of the solution method are referenced to the author’s PhD thesis.

Application of the model:

From the stated assumptions, the model can be used for any configuration using shell-and-tube heat exchangers. However, the range in which this model can be used is limited by the assumption of constant volume in the condensing zone. Also, since the saturated properties are assumed to be pressure independent, this model can only be applied for situations where the pressure changes are small, such as in small load changes. Start-up or shut-down behavior cannot be predicted with this model.

Discussion:

The model was validated against an existing set-up for one condition, which was a step change in the condenser water flow rate, keeping all other operating conditions constant. One simulation run was made by dropping down the water flow rate, and one by a step rise in the water flow rate. The response of the model was found to match that of the system, with some difference in the response rate of the compressor discharge temperature, which is explained as due to the neglected thermal mass of the compressor.

References:

Svensson M.C., 1994, “Studies on on-line optimizing control with application to a heat pump”, PhD Thesis, The University of Trondheim.

5.1.15 Researches on dynamic simulation and optimization of automobile air-conditioning system

Part 1: Mathematical model and dynamic simulation of automobile air conditioning system
Xuejun Sun, Weihua Liu, Xiongcai Que & Zhijiu Chen, 1999
Proc. 20th International Congress of Refrigeration (Sydney), Paper No. 577

Equipment Description:

The equipment modeled here is a HFC134a-based vapor compression air-conditioning system for an automobile. The model includes the heat exchangers, the throttling device, a receiver, an accumulator, the compressor and the passenger compartment. The evaporator is a direct-expansion type, the condenser is air-cooled, the compressor is reciprocating type and the expansion device is a thermostatic expansion valve or an orifice tube. The focus of this presentation is to study the effects on the passenger compartment, of variations in thermal load on the passenger compartment.

Purpose

This model was developed as a design and analysis tool for automobile air-conditioning system designers.

Assumptions:

- Pressure drops in the heat exchangers are neglected
- Flow in the heat exchangers is 1D homogenous
- Compression is polytropic
- Heat exchanger walls have zero resistance and finite capacitance
- Superheated and subcooled phases are treated as having void fractions of 1 and 0, respectively.

Mathematical Description:

The heat exchangers are modeled using a moving-boundary approach with each zone (liquid-phase, two-phase & superheated-phase) modeled in the lumped parameter form. Equations of mass and energy conservation are applied to each phase along with the heat exchange equations through the heat exchanger wall to the secondary fluid (air).

The equations for the refrigerant side use the void fraction concept for each zone, treating the liquid zone as having a void-fraction of 0 and the superheated zone as having a void-fraction of 1. The length of each zone is inherent in the mass and energy conservation equations for that zone.

The thermostatic expansion valve is modeled using a thermal-inertia delay for the sensing bulb. The sensed superheat is translated into a pressure signal, which is applied to the diaphragm force balance to determine the nozzle area. This nozzle area and the pressure difference across the valve are used to determine the mass flow rate.

The orifice tube is modeled by the orifice equation and has no 'input' signals from the evaporator.

The system modeled includes both a receiver and an accumulator, since either (but not both) can exist on any given system. In both these components, the refrigerant is modeled in its liquid and vapor phases as lumped parameters, in thermal equilibrium, and exchanging mass and energy of the phase-change. In both these components, both phases also exchange heat with the components body. Heat transfer from the body to the ambient is not included.

In the compressor model, individual strokes are not considered and the complete compressor model consists of a mass balance between suction and discharge and an energy balance between the polytropic work of compression, enthalpy rise of the refrigerant, thermal capacitance of the compressor wall (lumped) and the heat transferred to the ambient from the compressor wall.

In addition to the system components, the passenger compartment is modeled using a discretized disturbance-response transfer function, with the disturbances being the environment temperature and radiant heat and the responses being the cooling load and the compartment temperature.

Solution Technique:

No details were presented about the solution technique. The simulation was executed by 'uncoupling' the system at the compressor inlet. By this, it appears that compressor inlet conditions were initially enforced and the system was then allowed to iteratively converge to a solution depending on the state of the passenger compartment cooling load.

Application of the model:

The model developed for the refrigeration system is built on first principles. However, the assumption that void fractions of 0 and 1 can be used to represent sub-cooled and superheated phases in the heat exchanger, indicates that the model is intended for dynamics within a limited range from the steady state. Also, despite being an automotive system model, the compressor RPM is considered constant, or equivalently, the vehicle speed is steady.

Discussion:

The model was validated against an automobile test, using a Santana car. The system was operated with R134a, the vehicle speed was kept constant, and the only disturbances were from the ambient conditions. Good correlation is reported between the simulation and the measurement.

References:

Hara J. et al, 1991, "Study on the refrigeration cycle of automotive air-conditioners", Transactions of the JAR, Vol. 8, No. 3., pp. 51-59

Lu Yanli & Wu Peiyi, 1992, "Simulation and analysis of refrigeration system of car air-conditioning system", Journal of Refrigeration, No. 1 (in Chinese)

5.1.16 Mechanistic Model of Centrifugal Chillers for HVAC System Dynamics

Wang S,. Wang J. and Burnett J., 2000
Proc. CIBSE A: Building Services Engineering Research and Technology, Vol. 21, No. 2, pp. 73-83.

Equipment Description

The equipment modeled here are a single-stage and a two-stage centrifugal liquid chiller, with water as the secondary fluid. The compressor has variable inlet-guide-vanes for capacity control and the motor is cooled by tapping refrigerant from the liquid line.

Purpose

This model is developed as a design, analysis and optimization tool for centrifugal liquid chiller systems.

Assumptions

The model uses the following major assumptions:

- Refrigerant dynamics neglected
- Heat exchanger material dynamics neglected
- Only water dynamics are considered
- Negligible thermal resistance in heat exchanger material
- Negligible heat losses to the ambient
- Negligible pressure drops in the plumbing
- Polytropic compression
- Isenthalpic expansion
- Evaporator exit condition is saturated vapor
- Condenser exit condition is saturated liquid

Additional assumptions for the two-stage compression are:

- Compression ratios of both stages are equal
- Compression ratios equal to expansion ratios

Mathematical Description

This system model couples a detailed, mechanistic, quasi-steady-state centrifugal compressor model to a pair of shell-tube heat exchangers, lumping all the dynamics of the system into the inlet and outlet water temperatures. The centrifugal compressor is modeled in extensive detail using the conservation equations of mass, energy and angular momentum and the appropriate velocity triangles across the inlet guide vanes and the impeller. The theoretical head accounts for the polytropic and hydrodynamic loss components. The inlet losses, flow friction losses and incidence losses are accounted for. Both heat exchangers are modeled by an effectiveness-NTU approach on the refrigerant side. The quasi-steady-state leaving water temperatures are used to drive first order time-delays on the leaving water-side to represent the dynamics of the chiller. Similar first order time-delays are built into the inlet water-side also to capture dynamic variations in return water temperatures. Both heat exchangers use empirically determined overall UA values correlated to the water flow rate and heat transfer rate.

Solution Technique

The model solution is by time-wise integration of the lumped capacitances of the water at evaporator and condenser inlet and outlet. At each time step, the quasi-steady-state of the compressor and the heat exchangers is determined from inputs of the water flow rates and temperatures. This results in the determination of the system pressures, motor power and capacities of the two heat exchangers. A system level energy balance is used as the convergence criterion. No details have been presented about the integration algorithm or the system solution methods.

Application of the model

The centrifugal compressor model is sufficiently general to be used to model other similar machines. This formulation can be adopted for a given centrifugal compressor if full-load and part-load performance maps are available. The heat exchanger models however are inadequate to provide any internal information and provide little scope for studying effects of faults in the system.

Discussion

Validation:

The model validation was done using field data from two installations, one with a single stage centrifugal chiller and the other with a two-stage one. Full-load and part-load performance data was taken from these set-ups and used by a parameter identification program (built as a pre-processor to the model) to identify the necessary parameters needed in the model. These were then used to reproduce the measured data. A comparison of measured and predicted transient behavior of compressor power and condenser water leaving temperature are provided during what appears to a load-change, but the nature of the load-change is not detailed.

Comments:

This model is among the closest in terms of configuration, to the search criterion driving this literature review. The model of the centrifugal compressor is indeed exhaustive and is well suited for a physical understanding of internal losses within the compressor and alternate control schemes. This model could, potentially, be used with more detailed models of heat exchangers (such as McArthur & Grald [1987] or Rossi & Braun [1999]) to build a more truly mechanistic system model. Also, the inclusion of mechanical and thermal dynamics in the compressor would be an interesting value addition.

5.1.17 Modeling of In-Situ Liquid Chillers

Browne M.H. and Bansal P.K., 2000

Proc. 16th International Refrigeration Conference at Purdue, pp. 425-432

Equipment Description

The systems modeled in this work are a single-screw chiller and a twin-screw chiller, including a cooling tower for the secondary fluid.

Purpose

This model was developed as a design tool.

Assumptions

- Negligible pressure drops throughout the system
- Homogenous refrigerant properties within components
- Isenthalpic expansion
- Uniform temperatures across and along tubes
- Compressor dynamics neglected
- Uniform refrigerant flow-rate throughout the system

Mathematical Description

This work presents two steady-state models and two transient system models. One transient model is a simple, physical model and the other is a neural-network model. The physical model ignores the refrigerant dynamics in the heat-exchangers and only accounts for the dynamics of the heat-exchanger tube material and the water. The refrigerant side is therefore modeled quasi-statically assuming a uniform refrigerant flow-rate throughout the system. The dynamics of the tube temperatures in the heat exchangers and leaving water temperatures are modeled treating all the tube material and water as single lumps. The compressor is modeled using a steady flow displacement equation and an energy balance, assuming isentropic compression. The isentropic efficiency is mapped to the refrigerant flow rate and suction pressure. A constant motor efficiency is used to compute the motor power consumption.

Solution Technique

The system of 1st order ordinary differential equations developed in the physical model are integrated forward in time using a Cash-Karp 5th order Runge-Kutta algorithm, which incorporates dynamic integration time-step sizing based on the truncation error. The refrigerant inventory is enforced as an approximation at the beginning of the simulation as an approximation, as the heat exchanger formulation used does not capture the refrigerant migration phenomenon.

Application of the model

The model is suitable for use in controls studies as it captures the necessary system level dynamics with simplicity in the formulation. The inability to track refrigerant movement through the system or within the heat exchangers makes this model un-useable for superheat and sub-cooling predictions.

Discussion

Validation:

The model developed was validated using data from on-site measurements of a 650kW single-screw and a 300kW twin-screw chiller during start-up and also during a load increase and a load decrease. The model predictions are shown to agree well with the measurements. No comment is made however regarding the execution time.

5.2 Components and sub-systems

5.2.1 A System Mean Void Fraction Model for Predicting Various Transient Phenomena Associated with Two-Phase Evaporating and Condensing Flows

G. L. Wedekind, B. L. Bhatt and B. T. Beck
International Journal of Multi-phase flow, Vol. 4, 1978, 97-114

Equipment Description:

The process of evaporating and condensing flows within tubes is modeled here.

Assumptions:

- Complete two-phase zone is within the heat exchanger, i.e. exit conditions are not two-phase.
- Random fluctuations negligible compared to deterministic changes.
- Averaged heat flux in the two-phase region is time-invariant.
- Viscous dissipation neglected.
- Longitudinal conduction neglected.
- Kinetic energy changes neglected.
- Specific enthalpies of liquid and vapor are saturated values and functions of the mean system pressure.
- Vapor density is constant in superheated region.

Mathematical Description:

Based on the assumption that the complete two-phase zone lies within the heat exchanger, the steady state momentum equation is used to determine the area mean void fraction profile along the length of the heat exchanger and this is then integrated over the length of the two-phase length to obtain the system mean void fraction. In this process, the length of the two-phase region is maintained as the time-dependant variable. Using the computed system mean void fraction thus computed and assuming an average constant heat flux in the two-phase region, mass and energy balances on the two-phase region are developed into 1^{st} order ordinary differential equations. Solution of these equations identifies the instantaneous locations of the two-phase boundaries in the heat exchanger. The inputs to this equation are the entering quality and the mass flow rate.

Solution Technique:

The differential equation obtained above is solved for different inlet flow functions of time, by simple integration.

Application of the model:

This model is applicable to any two-phase heat-exchanger. However, it is limited to conditions where the exit condition is beyond two-phase. As such, the model can be used for steady-state, and some transient behavior, but not for the very early stages of start-up when exit conditions are still two-phase.

Discussion:

Validation:

This model has been validated using experimental data of two-phase flow in a 9m long, serpentine glass tube with straight sections between bends. In the evaporating mode, the model was used to predict the outlet flow-rate and the phase transition point when an exponentially time-varying inlet flow rate was imposed on the experimental test section. In the condensing mode, the model predicted the outlet flow-rate when the inlet flow-rate was increased and decreased sharply. In both modes, the model predictions were found to match the mean values of the measurements quite well.

Comments:

The development of the model is very clearly explained. The application of this method is also quite popular in the literature and this paper has been referenced widely. By avoiding the transient form of the momentum equation, significant reduction in complexity is achieved.

References:

Wedekind G.L. & Stoecker W.F., 1968, "Theoretical model for predicting the transient response of the mixture-vapor transition point in horizontal evaporating flow.", Journal of Heat Transfer, Vol. 90, pp. 165-174.

Wedekind G.L., 1965, "Transient response of the mixture-vapor transition point in two phase horizontal evaporating flow.", PhD Thesis, University of Illinois, pp. 63-70.

Fujie H., 1964, "A relation between steam quality and void fraction in two-phase flow.", A.I.Ch.E. Journal, Vol. 10, pp. 227-232.

Levy S., 1960, "Steam-slip-theoretical prediction from momentum model.", Journal of Heat Transfer, Vol. 82, pp. 113-124.

Zivi S.M., 1964, "Estimation of steady-state steam void fraction by means of the principle of minimum entropy production.", Journal of Heat Transfer, Vol. 86, pp. 247-252.

5.2.2 Mathematical Modeling Of A Direct Expansion Ground-Coupled Heat Pump System

P. Safemazandarani, J.A. Edwards, R.R. Johnson & Y. Mohammad-zadeh, 1990
ASHRAE Transactions, Vol. 96, Part 1, pp. 583-589

Equipment Description:

The equipment modeled here is a ground coupled heat exchanger (GLHXC) as part of a reversible vapor compression system. The focus is on the performance of this heat exchanger in transient operation in heating and cooling modes. The GLHXC consists of a refrigerant-in-tube heat exchanger submerged in a water bath under the ground.

Assumptions:

- Uniform thermal conductivity of the ground
- Two dimensional heat transfer in the water bath
- Uniform mixing temperature of the water

Mathematical Description:

The model for the GLHXC is developed from first principles, independently for when the system is operating in heating and cooling modes. In the heating mode, when the GLHXC works as an evaporator, the formation of ice and the rate of such ice formation is tracked by the addition of the ice thickness on the surface of the coil, in the energy balance. Ice formation on the coil is assumed to occur only at 32^{o}F at all times. The heat exchanger is an effectiveness-NTU model. The complete system model is built by replacing the model for the GLHXC, developed here, into an existing model for an air-to-air heat pump system, the development and validation for which is referenced to other work by the 1st author.

Solution Technique:

The solution is executed within a FORTRAN program, with the system operating in space cooling and space heating modes. The solution is iterative between the two-dimensional heat diffusion equation for the ground, the energy balance over the water bath vessel, energy balance of the water and energy balance of the heat exchanger. The convergence check is made on the total energy balance. From the solution, the saturation temperature in the heat exchanger, the length of the two-phase zone and the degree of superheat are obtained. A similar strategy is adopted for when the system is operated in the cooling mode, except that ice formation is not relevant.

Application of the model:

The model for the GLHXC is independent of the geometry, but the configuration is built into the energy balance equations. Also, since this model is grafted onto an external model for the indoor components, the extendibility of this model is not clear. As such the model appears to be unique for the specific equipment.

Discussion:

Validation:

The GLHXC model was validated against an existing ground coupled system and found to match the steady state behavior fairly well. However, transient predictions were off and are explained as due to differences in the boundary conditions between the simulation and the measurement.

Comments:

The model developed is fairly simple since the refrigerant dynamics were not considered. Also, there are significant deviations from the measurements in the transient region. This indicates that the model is only appropriate for steady state simulation or large scale transients (of the order of 1 day) which is the time scale on which the ground responds.

References:

Johnson R.R., Mohammed-zadeh Y, Edwards J.A. & Safemazandarani P., 1988, “Experimental evaluation of three ground-coupled heat pump systems”, ASHRAE Transactions, Vol. 94, Part 1.
Fischer S.K. & Rice C.K., 1981, “A steady-state computer design model for air-to-air heat pumps”, Report ORNL/CON-80.

Edwards J.A. & Pradeep V., 1985, “Heat transfer from earth-coupled heat exchangers: experimental and analytical results”, ASHRAE Transactions, Vol. 91, Part 2.

5.2.3 A General Dynamic Simulation Model For Evaporators And Condensers In Refrigeration:

Part 1: Moving Boundary Formulation Of Two-Phase Flows With Heat Exchange
Part 2: Simulation And Control Of An Evaporator
M. Williatzen, N.B.O.L Petit, L. Ploug-Sorensen, 1998
International Journal of Refrigeration, Vol. 21, No. 5, pp. 398-403

Equipment Description:

The equipment modeled in this work are the heat exchangers. The evaporator and condenser are treated equivalently.

Assumptions:

- 1D flow through a horizontal circular pipe of constant cross section
- Pressure drops are neglected
- Viscous losses and axial conduction through the refrigerant are neglected
- Pure refrigerants only
- A mean void fraction in the two- phase region

Mathematical Description:

The general forms of the mass, momentum and energy conservation equations are presented and reduced using the above assumptions. The momentum equation drops out completely.

The heat exchanger is treated in three zones – liquid, two-phase and vapor only regions. The mass and energy equations are written for each zone, introducing the lengths of each zone as additional variables. A third equation for each zone is obtained from the heat balance to the secondary fluid. This process yields a set of 3 coupled equations for each zone and 3 sets of such equations for all the three zones together.

These equations are algebraically re-worked, introducing a number of property related coefficients, to convert the implicit form of the time-derivatives of the state variables to the explicit form. This is done to simplify the subsequent implementation.

Solution Technique:

The model for the heat exchanger is applied to an evaporator. Four modes of evaporator operation are treated namely: liquid-two-phase-vapor, liquid-two-phase, two-phase and two-phase-vapor. For each mode, a module of equations for the combination of zones is coded. Details of the internal solution of each module are not presented. The focus in this work is on the handling of:

- occurrence of events such as change of operation mode and
- potential sources of numerical instability during change of mode.

The time step for the numerical integration of the equations is dynamically adjusted to smaller sizes when a mode change is imminent or anticipated and to larger sizes away from a mode change.

The solution is handled in an equation solver environment (which is not identified) capable of handling differential-algebraic-equations. A master program does the state-point bookkeeping and event handling that switches between the different modes of operation.

Application of the model:

The model has been developed for refrigerant in-tube heat exchangers and applies equivalently to evaporators and condensers, and in the general case, even to single phase heat exchangers. The execution of this model however, has to be within an equation solver environment capable of handling changes in equation structures.

Discussion:

Validation:

No experimental validation is presented. The model was run under certain common situations encountered in a vapor compression refrigeration system and the output behaved as expected.

Comments:

The development of the model from first principles is well presented. However, no clarifications are made on the solution of the equations of each zone. Also, it was difficult to follow the algebraic restructuring done for converting the implicit form of the equations to the explicit form.

References:

SINDA/FLUINT (Systems Improved Numerical Differencing Analyzer and Fluid Integrator) Version 3.2, Cullimore and Ring Technologies, Inc.

EASY5 Vapor Cycle System Design and Analysis, Boeing Computer Services.

Mac Arthur J.W. & Grald E.W., 1989, "Unsteady compressible two-phase flow model for predicting cyclic heat pump performance and a comparison with experimental data", International Journal of Refrigeration, Vol. 12, pp. 29-41.

Grald E.W. & MacArthur J.W., 1992, "A moving boundary formulation for modelling time-dependant two-phase flows." International Journal of Heat and Fluid Flow, Vol. 13, No. 3, pp. 266-272.

He X.D., Liu S & Asada H., 1994, "A moving interface model of two-phase flow heat exchanger dynamics for control of vapor compression cycle heat pump and refrigeration systems design, analysis and applications.", AES Vol. 32, ASME.

He X.D., Liu S & Asada H., 1986, "Modeling of vapor compression cycles for advanced controls in HVAC systems", Proc. of the American Control Conference, Seattle, Washington.

Wedekind G.L., Bhatt B.L. Beck B.T., 1978, "A system mean void fraction model for predicting various transient phenomena associated with two phase evaporating and condensing flows.", International Journal of Multiphase Flow, Vol. 4, pp. 97-114.

5.2.4 Dynamic Simulation Of Superheat At The Evaporator Outlet Of The Air-Conditioner With Inverter

Jing Xia, Xingxi Zhou, Xinqiao Jin & Zifeng Zhou, 1999
Proc. 20th International Congress of Refrigeration (Sydney), Paper No. 561

Equipment Description:

A detailed model of a fin-tube evaporator is presented, with the focus on studying the dynamic response of the superheat. The evaporator control is by an electronic expansion valve.

Assumptions:

- Refrigerant flow is 1D annular
- Refrigerant vapor and liquid are incompressible
- Refrigerant vapor and liquid are in thermal equilibrium
- Constant fin efficiency
- Tube and fin resistance is negligible

Mathematical Description:

The evaporator model is built in three zones, namely, the refrigerant side, the tube and fin and the air-side. On the refrigerant side, the above mentioned assumptions are applied and a set of 7 partial differential equations are obtained for the mass, momentum and energy conservation of the liquid and vapor phases and for the inter-phase mass balance. The tube and fin are modeled as a lumped capacitance with no resistance. It is not clear whether the liquid and vapor phases were discretized to a finite difference form, or they were treated as lumps.

The electronic expansion valve is modeled as an orifice with variable area and the compressor is modeled based on its volumetric efficiency.

Solution Technique:

The method of solution is not described at all and is also not apparent from the mathematical formulation.

Application of the model:

The purpose of this model is to study the dynamic response of the evaporator, this response being characterized by the superheat at the evaporator exit, under different perturbations. From the simulation, the static gain and time constant of the evaporator are determined.

Discussion:

The model developed was validated against an experimental plant built for the purpose. The simulation runs were made for changes in refrigerant flow rate, different air-flow rates across the evaporator, conditioned space temperatures, compressor speeds, condenser pressures and openings of the electronic expansion valve. The validity of the model was established to within 20% of the measurements.

References:

Wang H. & Touber S., 1991, "Distributed and non-steady-state modeling of an air-cooler", International Journal of Refrigeration, Vol. 12.
Guo Q.T., 1982, "Designing manual of refrigeration engineering", Chinese Construction Industrial Press, Beijing, PRC.

5.2.5 Experimental Evaluation Of The Use Of Homogeneous And Slip-Flow Two-Phase Dynamic Models In Evaporator Modelling

Jakobsen A., Antonius J & Høgaard Knudsen H.J., 1999
Proc. 20th International Congress of Refrigeration (Sydney), Paper No. 135

Equipment Description:

The equipment modeled here is a simple co-axial counter-flow evaporator. The focus of this work is on the relative merits of the slip-flow and homogenous flow models for evaporators.

Assumptions:

- Heat transfer is only between the refrigerant and the secondary fluid.
- Axial conduction is negligible in the refrigerant and the tube wall.

Mathematical Description:

The model development was done in SINDA/FLUINT using the transient forms of the mass, energy and momentum conservation equations all expressed in terms of the void fraction and saturated refrigerant (liquid and vapor) densities. The void fraction is expressed in terms of the slip and the refrigerant quality; homogenous flow is treated as a special case of slip-flow, with a slip of 1. The secondary fluid inlet condition is kept constant and the refrigerant entry and exit conditions are imposed on the evaporator as dynamic boundary conditions.

Solution Technique:

The solution of the model is executed within SINDA/FLUINT which uses a lumped parameter, finite difference approximation. Additional details of the solution process are not presented in this work. However, details of SINDA/FLUINT are referenced to Ploug-Sorensen et al [1997] (reviewed elsewhere in this report) and the on-line manual.

Application of the model:

The model developed is for the purpose of studying slip-flow vs. homogenous formulations, and as such, applies to refrigerant-in-tube heat exchangers. Also, the model is studied only for the case when the entry is two-phase. Although the governing equations are not constrained, applicability of this model to situations encountered in start-up or shutdown when the evaporator is completely flooded with liquid is not obvious.

Discussion:

Validation:

The model developed has been validated against an experimental test rig, with R22 as the refrigerant and ethanol as the secondary fluid. The validation has been done against two specific transient conditions namely, a sharp reduction in refrigerant flow into the evaporator and a sharp increase in the refrigerant flow out of the evaporator. The former was achieved by suddenly reducing the expansion valve flow-area and the latter was achieved by a sudden increase in the capacity control of the compressor. In both cases, the predicted transient flow-rates through the evaporator using the slip-flow and the homogenous flow model are compared with the measured flow.

Comments:

The work reported gives a very valuable insight into one of the most common assumptions regarding two-phase refrigerant flow in heat exchangers. The homogenous flow model is reported to be significantly inaccurate as compared to the slip-flow model and this inaccuracy is attributed to the large difference in the estimates of the refrigerant-mass as determined by the two approaches. The homogenous model predicts too quick a response in the evaporator because its prediction of the refrigerant charge is far too low. The slip flow model estimates a more accurate charge quantity and therefore evaporator response.

References:

Wallis Graham B., 1969, "One-dimensional two-phase flow", McGraw-Hill.

Antonius Jesper, 1998, “Distribueerede fordampermodeller pa flere detaljeringsniveauer”, MS Thesis, Department of Energy Engineering, Technological University of Denmark.
Ploug-Sorensen L, Fredsted J.P. & Willatzen M, 1997, “Improvements in modeling and simulation of refrigerant systems: Aerospace tools applied to a domestic refrigerator”, International Journal of HVAC&R Research, Vol. 3, No. 4.

5.3 Applications

5.3.1 A Dynamical Model Adequate For Controlling The Evaporator Of A Heat Pump

József Nyers & Gisbert Stoyan, 1994
International Journal of Refrigeration, Vol. 17, No.2, pp. 101-108

Equipment Description:

The equipment modeled here is the evaporator alone as part of a refrigeration system. The focus is on the response of the evaporator to system controls. The compressor and the expansion valve are also included, albeit simplistically.

Assumptions:

- Refrigerant flow is distributed equally through all the pipes
- Refrigerant flow is homogenous 1 dimensional
- Thermal resistance of the tube wall is negligible
- Axial conduction in the tube wall is negligible
- Refrigerant is in thermal equilibrium in the 2 phase region

Mathematical Description:

The above assumptions are applied to the general forms of the equations of mass, momentum and energy conservation in transient form to obtain linear first order partial differential equations in time and the co-ordinate along the tube length. These are coupled to the heat transfer equations. The model is a moving boundary formulation and considers only the two-phase and the superheated phase. The interface is tracked by monitoring the point at which the quality is 1. The evaporating phase, the interface and the superheated phase are modeled by including the appropriate forms of equations of state for two-phase and superheated conditions.

The expansion valve and compressor are simplistically modeled using algebraic equations with lumped parameters.

The model operates on 5 independent variables, which are treated as control parameters. These are the temperature and velocity of the chilled water, throttle coefficient of the expansion valve, compressor speed and condenser pressure.

Solution Technique:

The conservation equations are re-written in a fully implicit finite difference formulation. The momentum and energy equations are backward differenced while the mass conservation equation is forward differenced. The explanation for this is not given. These are then combined with the refrigerant property equations to give sets of 6 equations for each of the 5 zones of interest, namely, the expansion valve, the two-phase region, the interface, the superheated region and the compressor. These equations are arranged in a tri-diagonal block-matrix and solved using the block-Gauss solution method. At each time step, this set of equations is solved, followed by solution of the heat transfer equations.

The simulation was executed under different control conditions. This consisted of varying each of the 5 independent variables in 3 different ways, namely, step jump, exponential saturation and periodic oscillation.

Application of the model:

This model was run using geometric data of an existing heat pump. However, validation experiments have not been reported. The model developed for the evaporator is applicable to any refrigerant-in-tube evaporator.

Discussion:

This model is among the few available in the literature where the transient form of the momentum equation is solved to determine pressure and temperature drop simultaneously along the evaporator.

References:

Nyers J. & Stoyan G, 1990, "Analysis of the dynamic model of the dry evaporator of heat pumps and refrigerators", Bull. Of Applied Mathematics, Vol. 683 (LIV), pp. 279-285.
Nyers J. Stoyan G & Baclija P., 1991, "On the numerical simulation of the evaporator of a heat pump with explicit determination of the phase border", Applied Mathematics, Veszprem.

5.3.2 Modeling of Vapor Compression Cycles for Multivariable Feedback Control of HVAC Systems

Xiandong He, Sheng Liu & Haruhiko H. Asada, 1997

ASME Journal of Dynamic Systems, Measurement and Control, Vol. 119, No. 2

Equipment Description:

The equipment modeled here is a complete vapor compression system consisting of the reciprocating compressor, the heat exchangers and the expansion device. The focus of this work is on building a model suitable for multi-variable feedback control.

Assumptions:

- Refrigerant flow in the heat exchangers is 1D
- Axial heat conduction is negligible
- Pressure drops in the heat exchangers are negligible
- Viscous friction is negligible
- Compression is polytropic

Mathematical Description:

The system model is built from first principles. The heat exchangers are modeled on the moving-boundary lumped parameter approach, using the time-invariant system mean-void fraction method of Wedekind [1978] to model the two-phase region. The assumption of negligible pressure drops eliminates the momentum equation and the different phase zones in the heat exchanger are represented using only dynamic mass and energy balances. Since zone-wise spatial averaging is presumed, the space-time partial derivatives reduced to time-based ordinary derivatives. The evaporator is characterized by 5 state variables, which are: two evaporator wall temperatures (one for each phase zone, i.e. two-phase and superheated), the evaporating pressure, the exit enthalpy of the refrigerant and the length of the two-phase zone. The condenser is similarly characterized by 6 state variables, which consist of the above 5 and a third wall temperature corresponding to the presence of a sub-cooled region.

The compressor is modeled using a volumetric efficiency and polytropic compression and the expansion valve is represented by an orifice equation without dynamics.

These individual component models are combined into a global set of equations as a 11^{th} order system, in matrix form.

Solution Technique:

The solution of the 11^{th} order matrix equation is not explained, but presumably follows a standard matrix-inversion technique. For the purpose of introducing multi-variable feedback control strategies, the model is mathematically manipulated further as follows.

As a first step, the inherent non-linearities of the system are eliminated by focusing attention on the small-scale transients associated with load variation, i.e., the dynamics of the system are studied as it deviates from a preset steady state point. After linearizing the model, a reduction of order is accomplished by the method proposed by Edgar [1975]. This consists essentially of studying the dominant dynamics of the system to step-input responses. Using this method, the model is reduced from a 11^{th} order to a 4^{th} order one.

Application of the model:

The model can be seen as developed in two stages. The first is the 11^{th} order matrix equation which is applicable to any basic vapor compression refrigeration system. The second is the linearization of the model which narrows the applicability to small-scale dynamics.

Discussion:

<u>Validation:</u>

The validation of the model was done at the first of the two stages mentioned above. This was done against a residential split-type air-conditioner rated at 2.9kW cooling capacity. The model is linearized around a known steady state condition of the experimental system, and the results of the model are compared with those of the test set-up, for a step increase in the compressor speed. The results are shown to compare well, while also indicating the possibility of order reduction.

References:

Edgar T.F., 1975, "Least squares model reduction using step response", International Journal of Control, Vol. 22, No. 2, pp. 261-270.
Wedekind G.L., Bhatt B.L. & Beck B.T., 1978, "A system mean void fraction model for predicting various transient phenomena aassociated with two-phase evaporating and condensing flows", International Journal of Multiphase Flow Vol. 4, pp. 97-114.

5.3.3 Numerical Prediction Of Dynamic Performance Of Vapor Compression Heat Pump Using New HFC Alternatives To HCFC-22

Sami S.M & Dahmani A, 1996

Journal of Applied Thermal Engineering, Vol. 16, Nos. 8/9 pp. 691-705

Equipment Description:

The equipment modeled here is an air-to-air reversible heat pump system consisting of a hermetic compressor, direct expansion evaporator, air-cooled condenser, thermostatic expansion valve and an accumulator. The focus of this work is to study transient behavior of the system using alternatives to HCFC-22; the alternatives considered are HFC-407a, HFC-507 and NARM-502 (a blend of R22, R23 and R152a).

Assumptions:

- Refrigerant flow in the compressor is compressible and the processes are isentropic or polytropic.
- Refrigerant vapor behaves as an ideal gas in the compressor
- Kinetic and potential energies of refrigerant during compressor strokes are negligible
- Liquid and vapor in the heat exchangers are in thermal equilibrium

Mathematical Description:

The model developed in this work is similar to that reported in Sami et al [1987] (reviewed elsewhere in this report). A significant enhancement is the use of finite differencing within the drift-flux model of the above. Also, since this work focuses on the working fluid and its effects on the system, the major differences are in accurate modeling of the refrigerant's thermo-physical and transport properties. However, for completeness, a brief review of the model is given here.

The system is modeled from the fundamental principles of mass, energy and momentum conservation applied to different control volumes within each component. In the heat exchanger, the liquid and vapor phases are treated as two different control volumes exchanging mass between each other, and energy to the heat exchanger wall. A similar analysis is applied to the accumulator. The compressor model is built along the lines of Dhar and Soedel [1978]. The properties of the refrigerant(s) are evaluated using REFPROP Version 4.01. Some (unclarified) numerical modifications to this are made necessitated by compatibility with the main refrigeration system model program. The boiling heat transfer coefficients are determined from a Bo Pierre's type of correlation developed for the refrigerants studied. The condensation heat transfer coefficient is referenced to Sami & Schnotale [1992]. Two phase mixture characteristics are determined from the Carnahan-Starling-Desantis equation of state.

Solution Technique:

The complete program was coded in FORTRAN IV and executed on a Pentium computer. The governing equations are solved for each control volume, by time-wise integration over the system, and internal convergence within each time-step. Details of the numerical techniques used for the solution are not provided.

Application of the model:

This refrigeration system model is developed from first principles and is therefore generic within the assumptions made. The complete model however is specific to the refrigerants for which the properties have been obtained from REFPROP, after the *numerical modifications* required for compatibility with this model. On a hardware level, this model is limited to the specific configuration that it is developed for. However, its applicability is enhanced by its interface with REFPROP and hence to any refrigerant, be it pure, azeotropic or non-azeotropic.

Discussion:

The model developed was validated against an experimental 3-ton air-to-air heat pump system from which measurements were taken for start-up, using NARM502 (a mixture of R-22, R-23 and R152a) as the refrigerant. This refrigerant was developed by the authors and proposed as a drop-in replacement for HCFC22, CFC12 and CFC502.

The start-up behavior was shown to match well with the measured data, except for the first 35s after start-up. This is attributed to the neglecting of changes in the refrigerant distribution along the evaporator. The model was also shown to predict system behavior when the evaporator air temperature was varied.

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